FOUR-STROKE PERFORMANCE TUNING

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A. GRAHAM BELL

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AIR DENSITY WITHIN THE ENGINE

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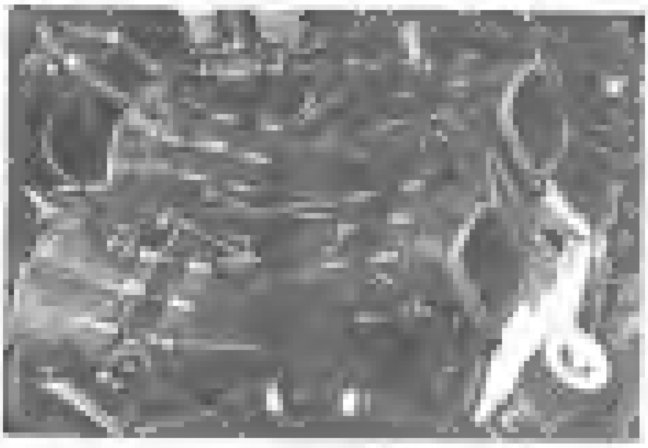

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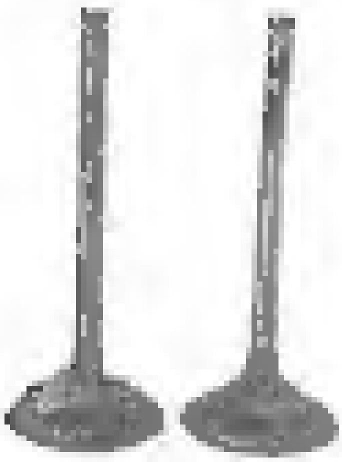
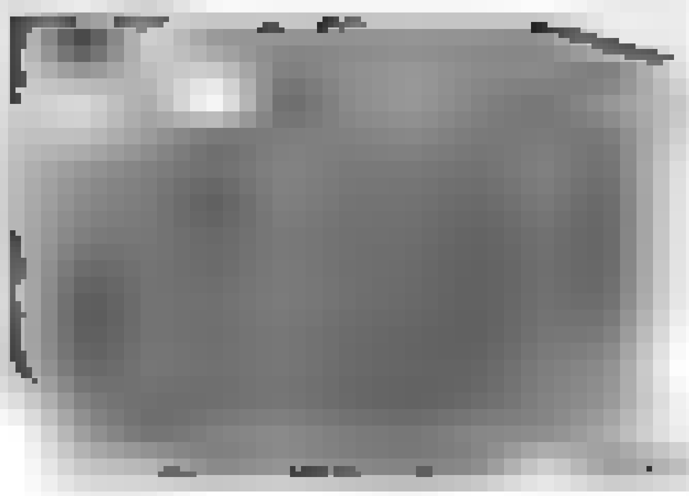
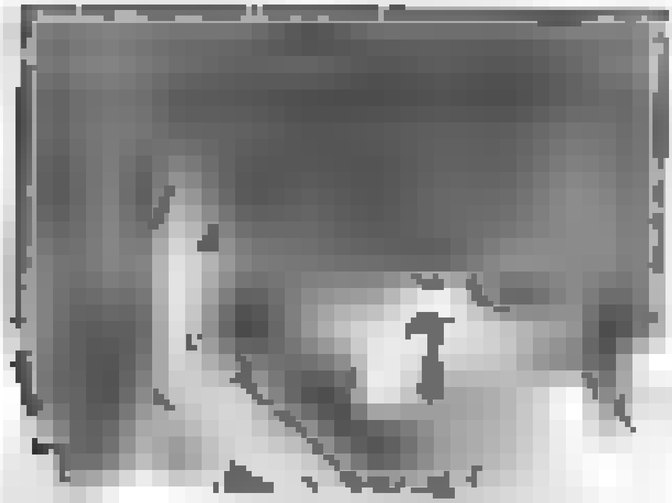
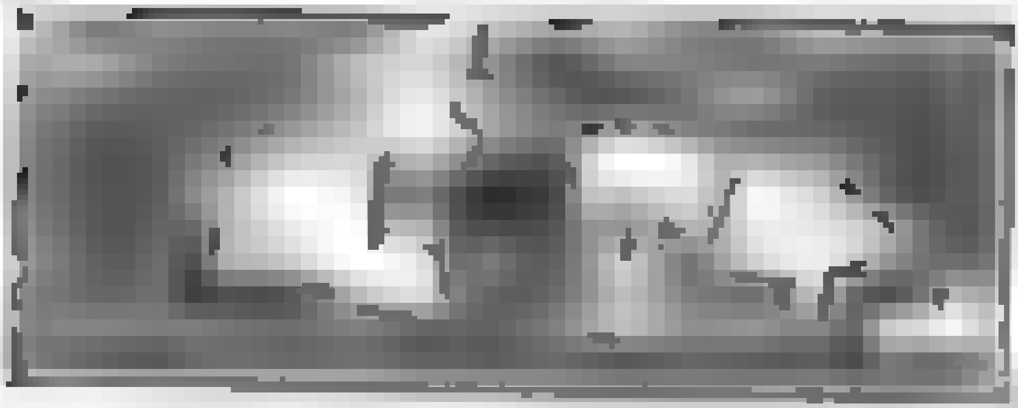
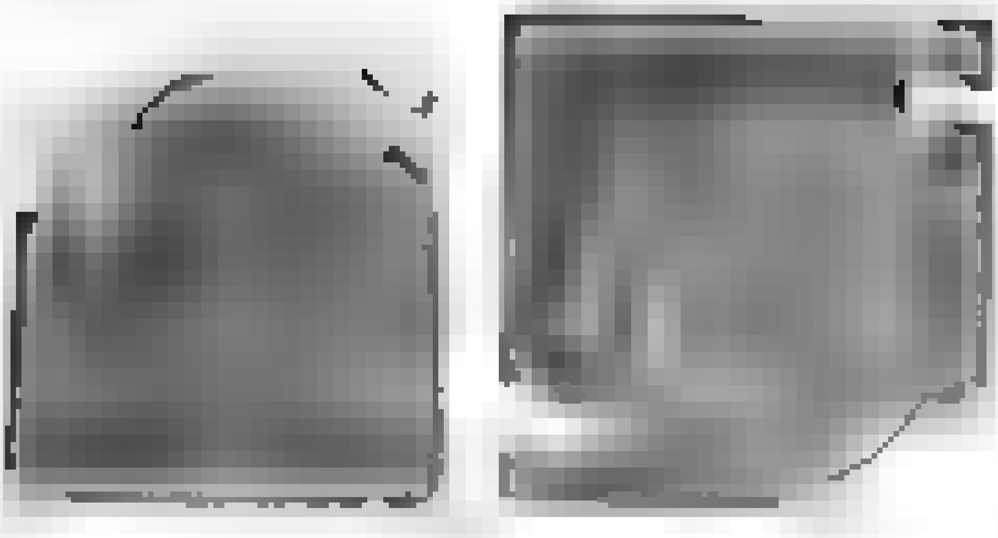
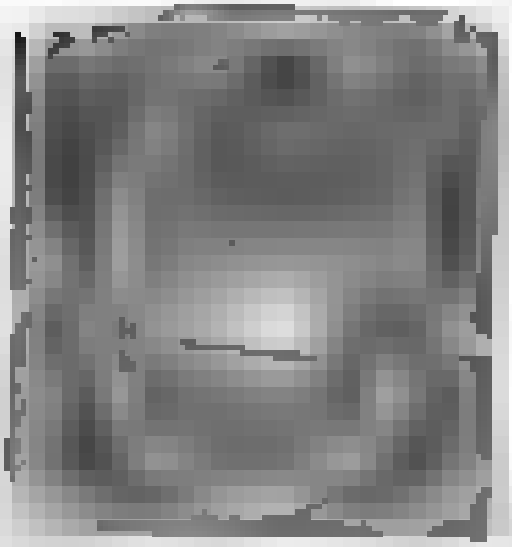


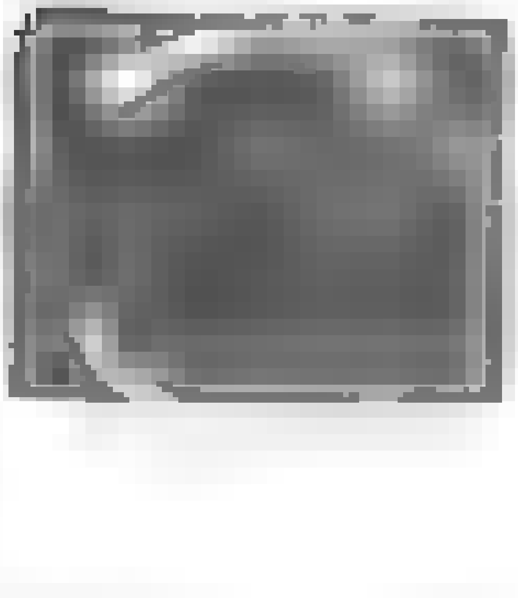


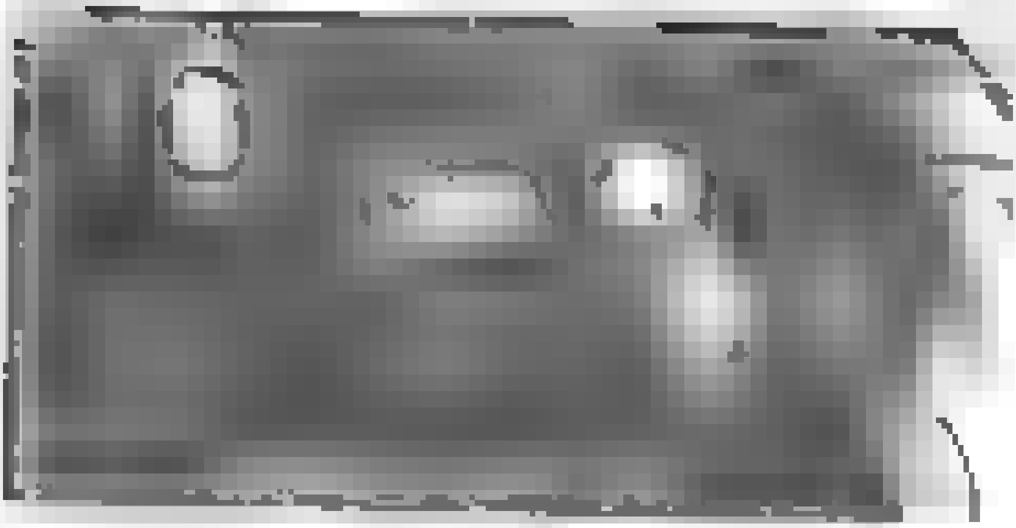


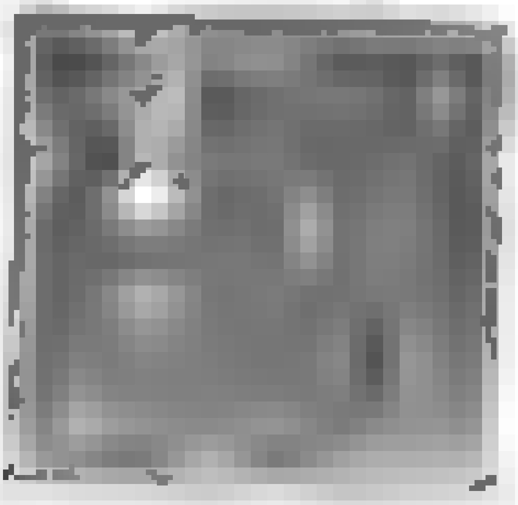


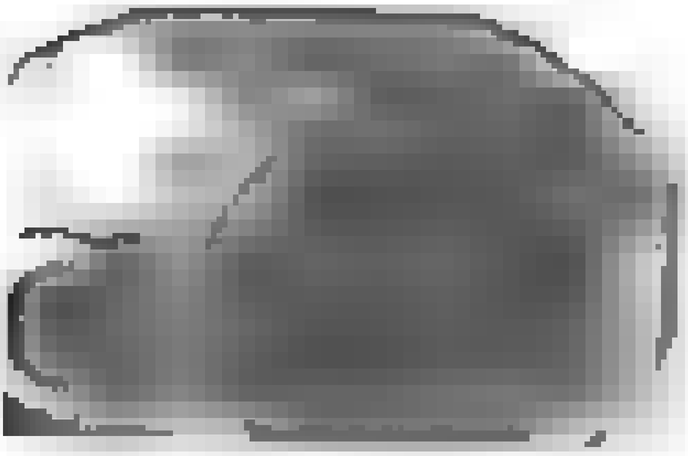


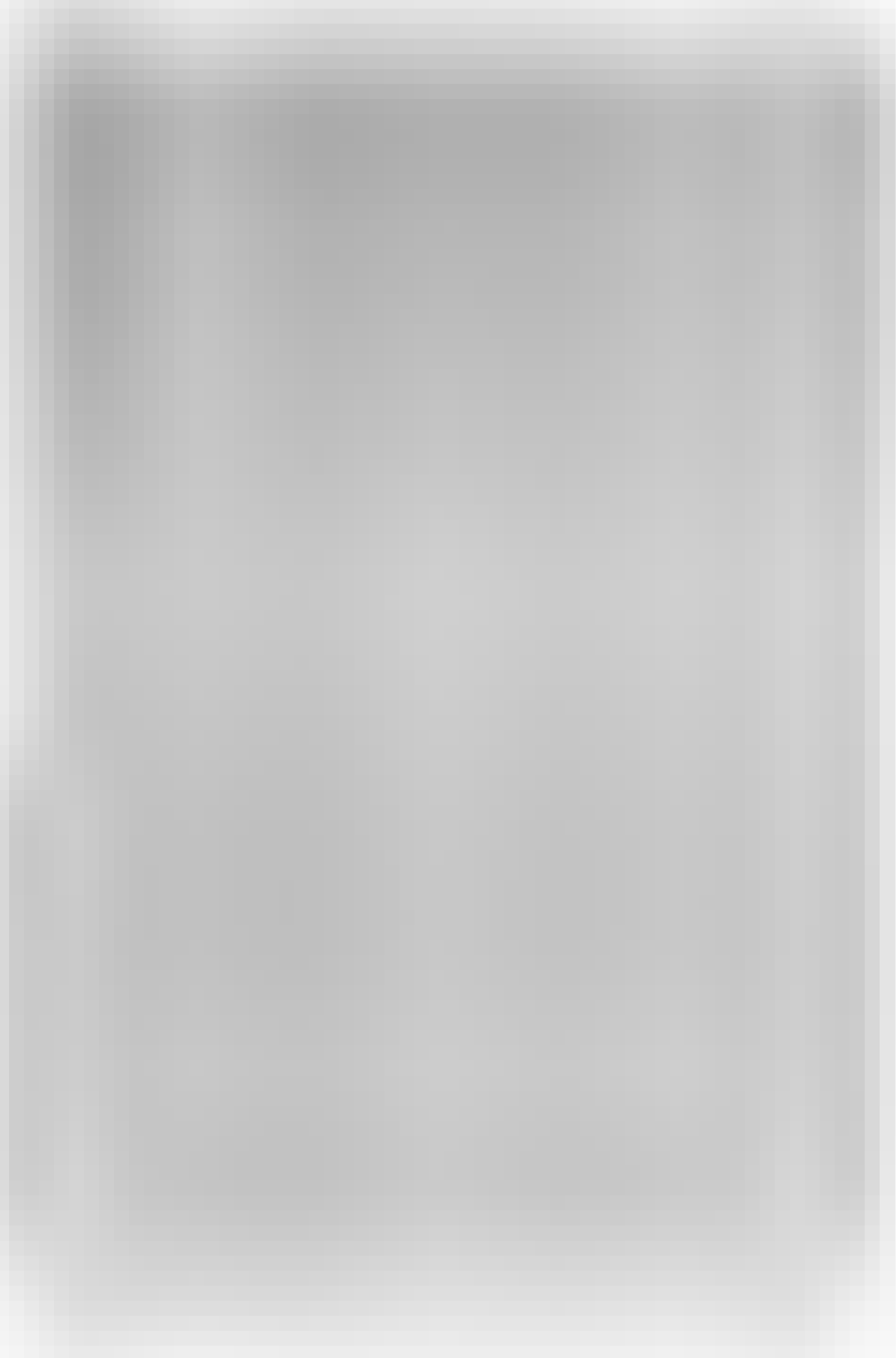












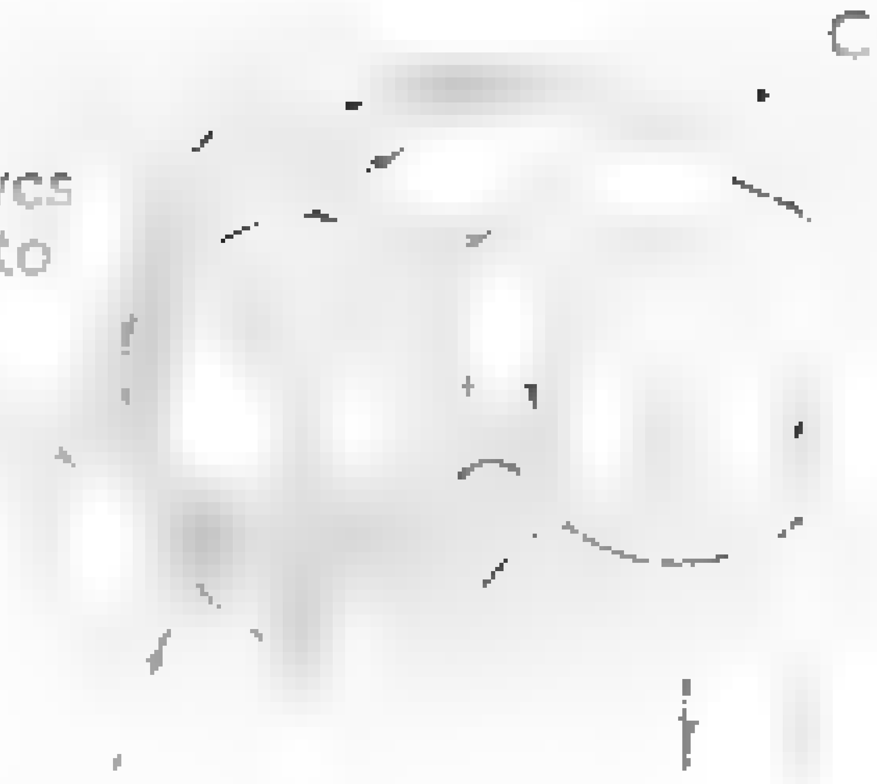


Figure 3.11 Austin Mini chamber modifications

Under certain conditions it is possible for the combustion flame to pre-heat the fuel charge directly in front of the flame to the point of self-ignition. When this flame collides with the spark-ignited combustion flame, explosion-like combustion results (detonation). The quench area normally prevents this by removing excess heat from the outer gases. With the temperature lowered, the end gases do not self-ignite to initiate a dangerous detonation condition.

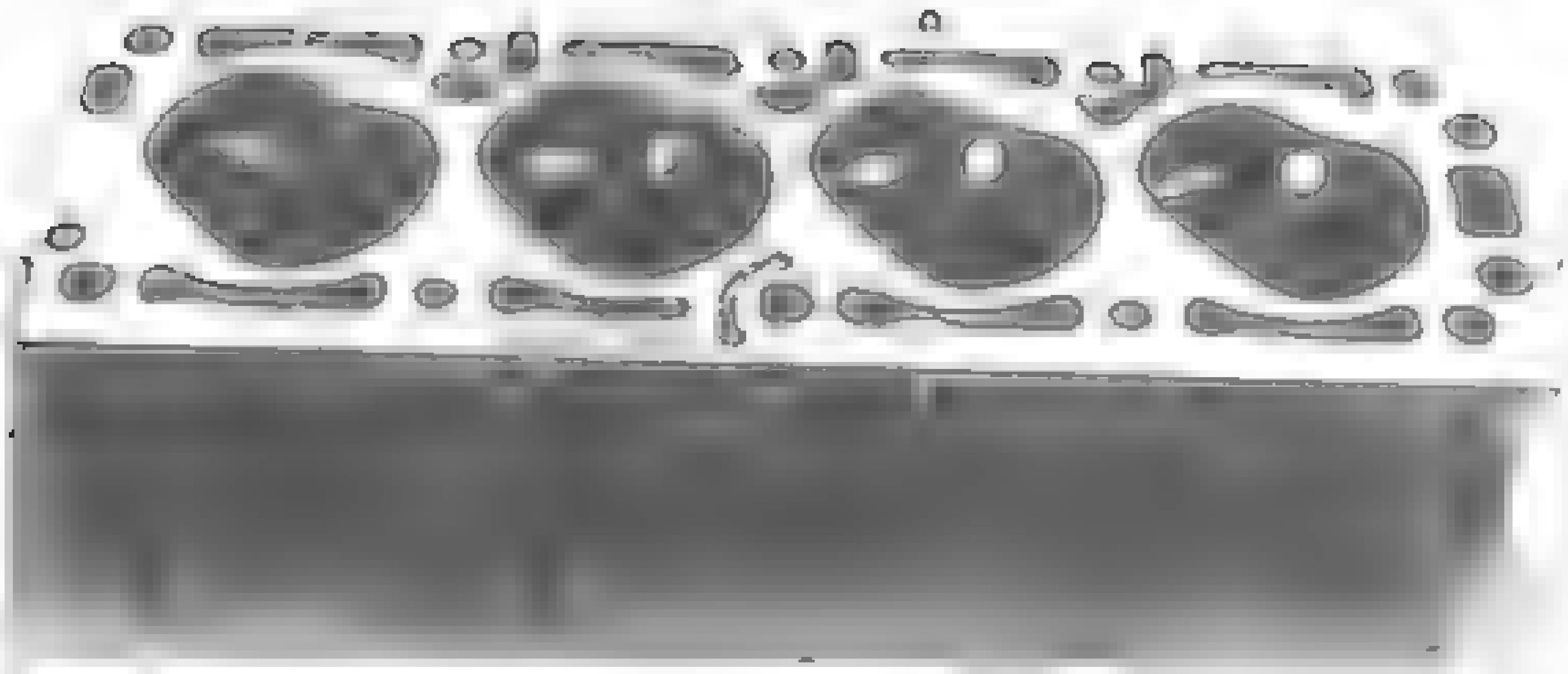
The quench area also lowers the piston crown temperature by momentarily restricting the combustion flame to just the area of the combustion chamber. This increases piston and ring life, and helps prevent the piston from becoming a hot spot, able to pre-ignite the fuel/air charge.

The exhaust valve requires very little unshrouding at all. I generally work to a figure of 60 % of valve lift for the radius between the valve head and chamber wall. It is a mistake to exceed this figure as the exhaust valve flows very well partially shrouded, in fact, it seems to enjoy being shrouded. I well remember an occasion where I was able to pick up 6hp on a VW 1600 by shrouding the exhaust valve a little. Power went up from 163 to 169hp, which represented a 4% gain.

The heart-shaped chamber of the Mini (Figure 3.11) is an easy one to modify. It yields high power due to the inlet port offset producing a good swirl effect. The valves should be unshrouded as with a bath-tub chamber.

THE TWISTED BATH-TUB CHAMBER

A later development of the bath tub chamber, which also borrows from the Austin Mini in that the inlet port enters the chamber at a similarly favourable angle, is what is called the 'twisted bath-tub' chamber. This chamber is found in the Isuzu-designed Opel/Holden/Vauxhall Family I and Family II OHC engines. As you will notice from the photograph, the combustion chamber is turned, or 'twisted', through 45°, producing a head with inlet ports on one side and exhaust ports down the other side. With the chamber twisted like this a much greater area around the periphery of both the inlet and exhaust valves is unmasked, so flows into and out of the engine are improved. Also there is a good squash area opposite the spark plug, which is itself located deep into the chamber, giving a good quick burn. For even better inlet flow



Twisting the combustion chambers opens up the inlet and exhaust flow path while maintaining generous squish areas

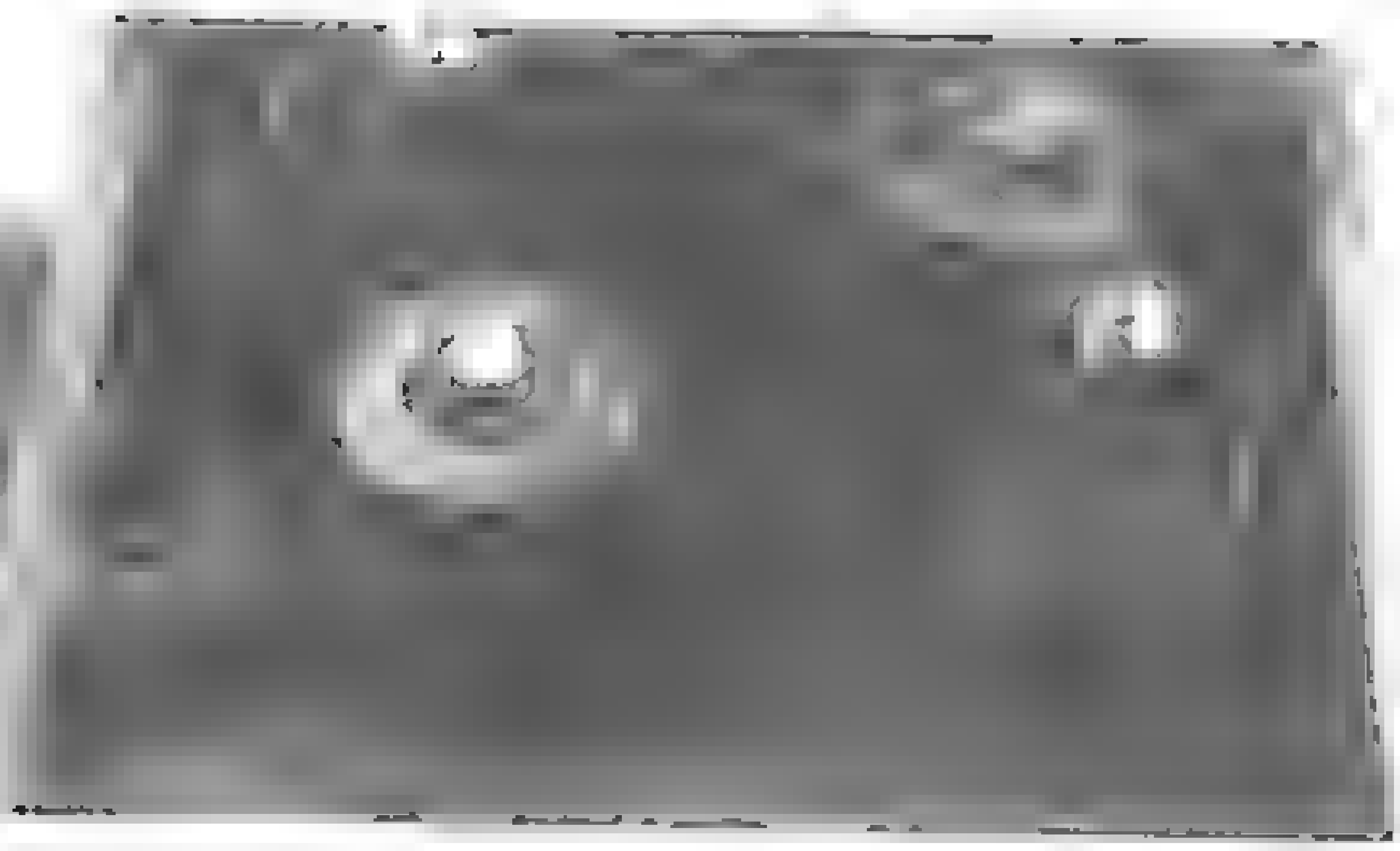
some of the squish area near the spark plug can be cut back toward the cylinder outline, but take care not to overdo this.

Remember that we were talking before about squish and its importance in the combustion process. The primary or major squish area should be somewhere about opposite the spark plug, to squish the bulk of the fuel/air mixture over into the combustion area. Now behind the spark plug we have a secondary squish area. Its effect is also to squish and create a pressure wave. As the spark plug ignites the mixture, this wave assists the flame front to travel through and ignite the mixture that the major squish area has pushed towards the spark plug.

If the combustion chamber is cut back right to the bore wall, what is the result? Gas flow may improve, but remember that I said right at the outset that gas flow has to be balanced against good combustion. With no secondary squish to assist the flame front and impede the mixture rushing across from the major squish area, you may lose power. This is what can happen: the mixture being purged across to the spark plug bangs into the chamber wall and spark plug and devaporises, wetting the chamber wall and possibly the spark plug too, depending on its temperature. Try holding your hand in front of an aerosol spray pack when you are spraying and you will see what I am [illegible].

[illegible] the very least, a head modified in this way will waste fuel and dilute your [illegible] oil. Some of that lovely fuel mixture you have managed to cram into the cylinder will end up going out of the exhaust or, at low revs and/or with a cold motor, will end up in the sump, wrecking your super-good oil.

One problem with many of these Opel heads is that the inlet ports are way oversize, so keep the grinder away from them or these engines become even more sluggish at lower rpm. Fortunately a number of models have good size valve [illegible] inserts and the valve spacing is such [illegible] s can be easily fitted. B[illegible] valves will provide a good hp increas[illegible] without taking anything away from the bottom end. In the later Gen II h[illegible] ds GM got it right with smaller but much superior flow inlet ports. This change boosted midrange power and also maximum power even though [illegible] milder cam with less lift and duration [illegible]



...g the valves improves flow b... ause the head of the opening valve moves aw... e wall of the combustion chamber and provides a wider flow path

The 2- and 2.3-litre Ford Pinto head appears to be the same ... the twisted bath tub but it is in fact slightly different in that the inlet and exhaust valves are not ver... As you can see from the photograph, both valves are tilted, or 'canted', a few d... Wh... this gives rise to all sorts of valve ... problems, as anyone who competes with these engines is only too well aware, splaying or canting the valves does have several benefits. First, as the valve opens it moves away from the combustion chamber wall thus opening up the flow path. This also opens up the way to fit larger valves, w... would otherwise be severely masked if arranged vertically. Additionally, as we will discuss in depth later, reducing the port-to-valve ... le improves flow around the v... head into the combustion chambe...

As with the early Gen I Opel, these heads also have overly generous inlet port relative to the standard-size inlet valves. Only when larger valves are fitted should the ports be touched.

THE WEDGE CHAMBER

The wedge chamber in its most basic form is in reality just a bath-tub c... inclined, usually at around 15°. This is the type of chamber found in almost eve... American V8 and it presents the serious tuner with many problems, particularly wh... the chamber is tilted steeply, as it is in the small-block Chev at 23...

Why use the wedge chamber then, you may ask? Figure 3.12 illustrates the m... obvious benefit from this design, namely the reduction of the bend in the inlet por... allowing the valve to flow better, particularly on the lower side. A straight (not ... ed) port flowing into a bath-tub chamber flows only about 20% of the mixture on ... er or 'dead' side of the valve ... the wedge chamber can flow 25–30% of the mixture on this side.

The other obvious benefit is the concentration of fuel/air mixture clos... plug in the deeper ... ctively shortens th...

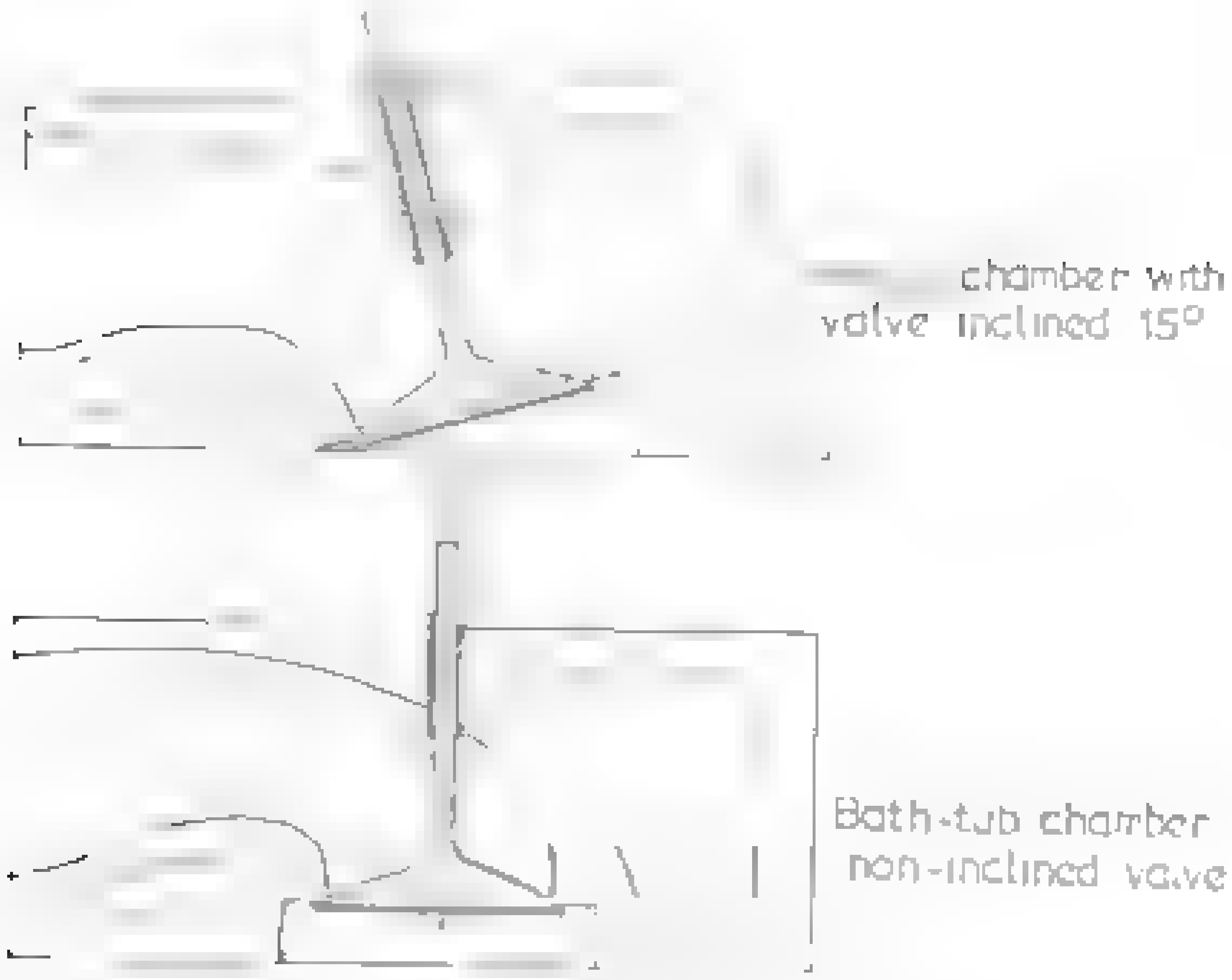


Figure 3.12 A wedge chamber decreases the port entry angle

However, and this again highlights the importance of balancing good air flow with good combustion, in performance and competition applications we soon run into a situation where with many wedge chambers we just cannot get better than a 9.5:1 compression ratio without resorting to high top pistons. Any piston becomes in effect [illegible] so loads on bearings and connecting rods increase. This could mean that heavier, more expensive rods and crank, or a lower rpm limit, are necessary.

This has led to a serious rethink in Detroit. First in 1989 Chevrolet introduced its 18° Bow Tie range of heads for competition engines. These heads feature smaller, [illegible] detonation trouble. This has meant less piston crown protruding into the combustion chamber while allowing the compression ratio to climb to 15:1 on 112 octane race fuel.

Then at the end of 1996 Chev introduced their revised small-block engine, dubbed the Gen III. While still retaining wedge chambers, the valve inclination is further reduced to 15° to produce a more compact combustion chamber and, like the LT1 [illegible]

Ford SVO has done something similar with their N351 cast iron competition heads. In these the valves are inclined at only 10° as opposed to the standard 22°. This effectively makes the combustion chamber more compact and means that only a small protrusion is needed on the pistons to achieve a high compression ratio. To restore air flow the valves were repositioned within the combustion chamber and the inlet ports were raised 0.3in.

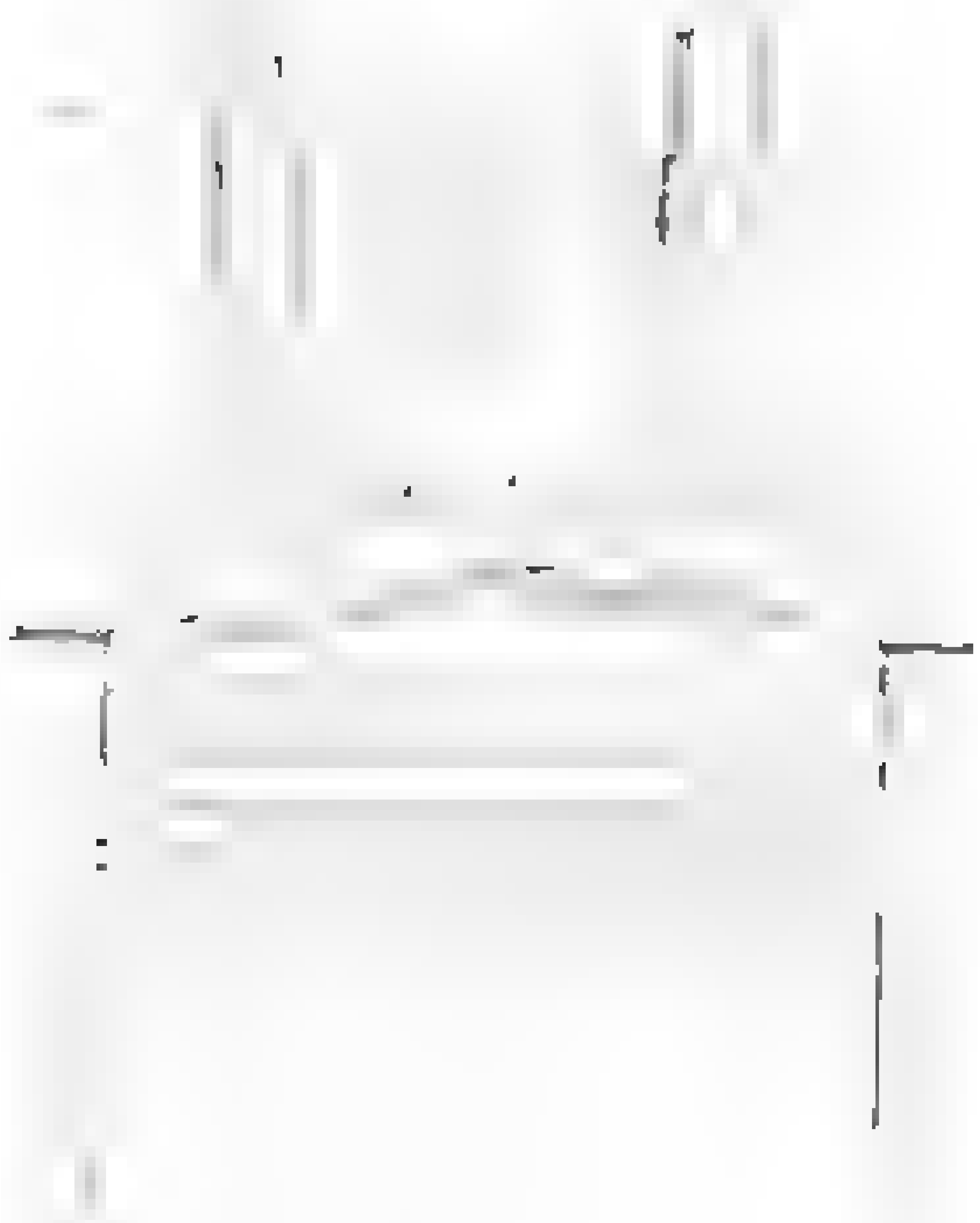


Figure 3.13 big-block Chev head design

CANTED-VALVE WEDGE CHAMBER

[illegible] chamber through 20°, although full advantage could not be taken of this because of 'good' and 'bad' inlet ports. When compared with the Ford Cleveland Boss 302/351 [illegible] why the Chev inlet ports are dubbed as being either 'good' or 'bad' (Figure 3.14).

Like the early design wedge heads, the Chev big block canted-valve wedge [illegible] ratio. Such a large compression lump causes all kinds of combustion problems. Even with a fairly decent cam, high-performance road engines run into detonation problems over 9:1 on 98 octane (Research octane) petrol, and even race engines will not handle anything over 12:1 on 110 octane race fuel. This is not to say that the canted-valve

Figure 3.14 Two high performance 'twisted wedge' chambers

wedge chamber cannot be made to work, it can, as Chev demonstrated when they produced competition heads for both their small-block V8 and 90° V6 with new-style compact canted valve wedge chambers. Like the Ford Cleveland these later heads have all 'good' symmetrical inlet ports, and while the inlet and exhaust valves are still canted [illegible] considerably to provide an inlet valve inclination of 16° and an exhaust valve inclination of 11° to [illegible]

open chamber Mark IV volume of nearly 120cc

THE HERON DESIGN

This leads us to what I regard as the most horrible design of all time, the Heron head [illegible] it is nothing like so good. The design utilises a flat face head without a combustion chamber. Instead, combustion takes place in a combustion chamber cast into the piston crown. Ford used [illegible]

motor, Chrysler also used it in their Avenger motors

Having no effective squish area, this design suffers chronic ignition problems [illegible] If you must stick with this design rather than using a head with combustion chambers machined into it [illegible]

rough. Machine the top of the cylinder block to bring the pistons to around 0.005-0.007in from the deck. This will induce some squish effect. I have found that recessing the valves about 0.060in into the head (Figure 3.15) improves not only the

which will improve the compression as well.

Besides the combustion problem, a basic mechanical problem exists with the

throttle response at lower rpm is not good.

Now, I do not deny that Cosworth used this design on their very successful 1-litre

rattle all it likes, so piston distortion and/or seizure would not be such a problem.

HEMI AND SEMI-HEMI CHAMBER

Decreasing the valve-to-port angle is really what high power outputs are all about. By coaxing the 'dead' side of the port and valve to flow better, we can cram more mixture

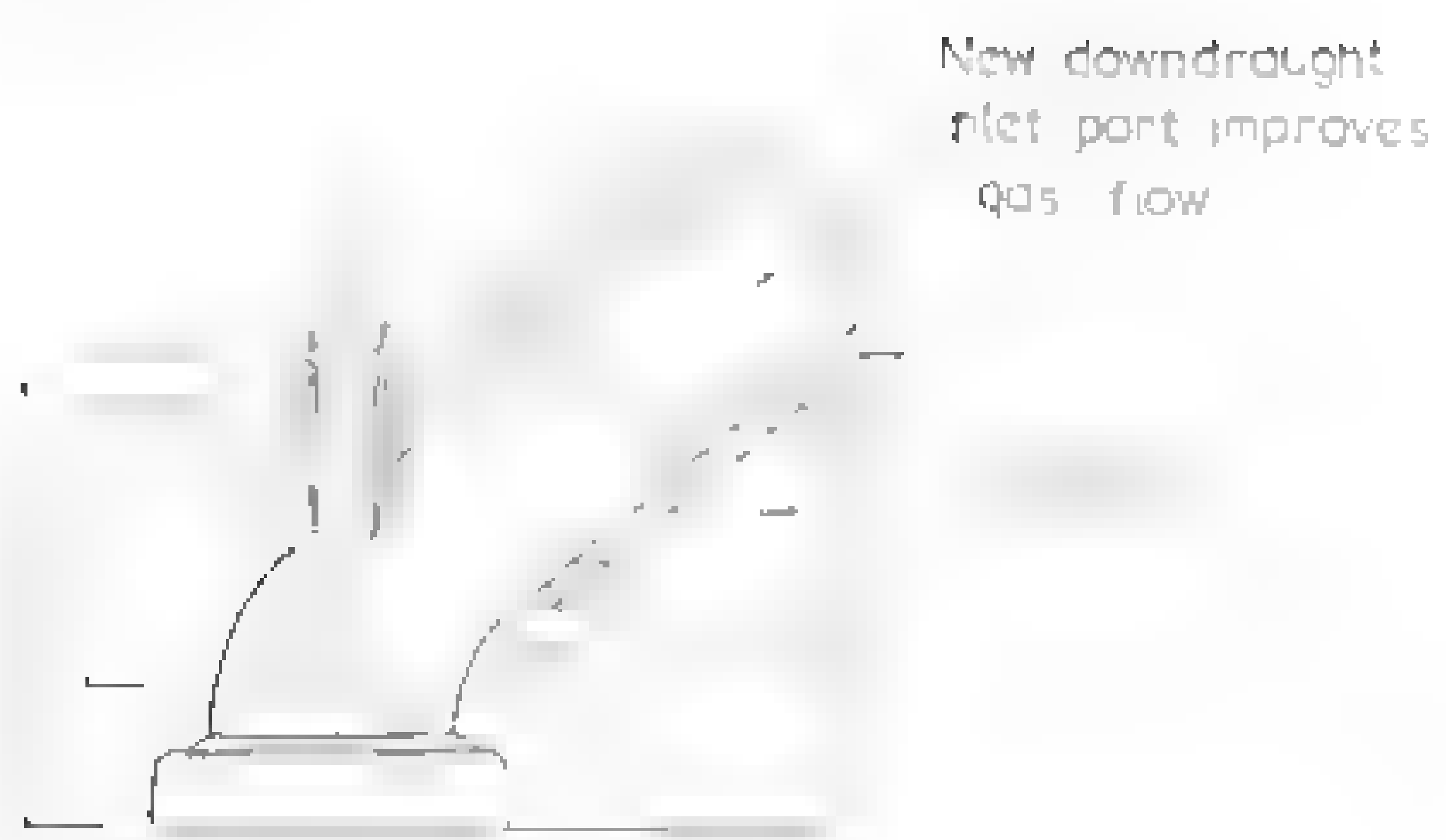
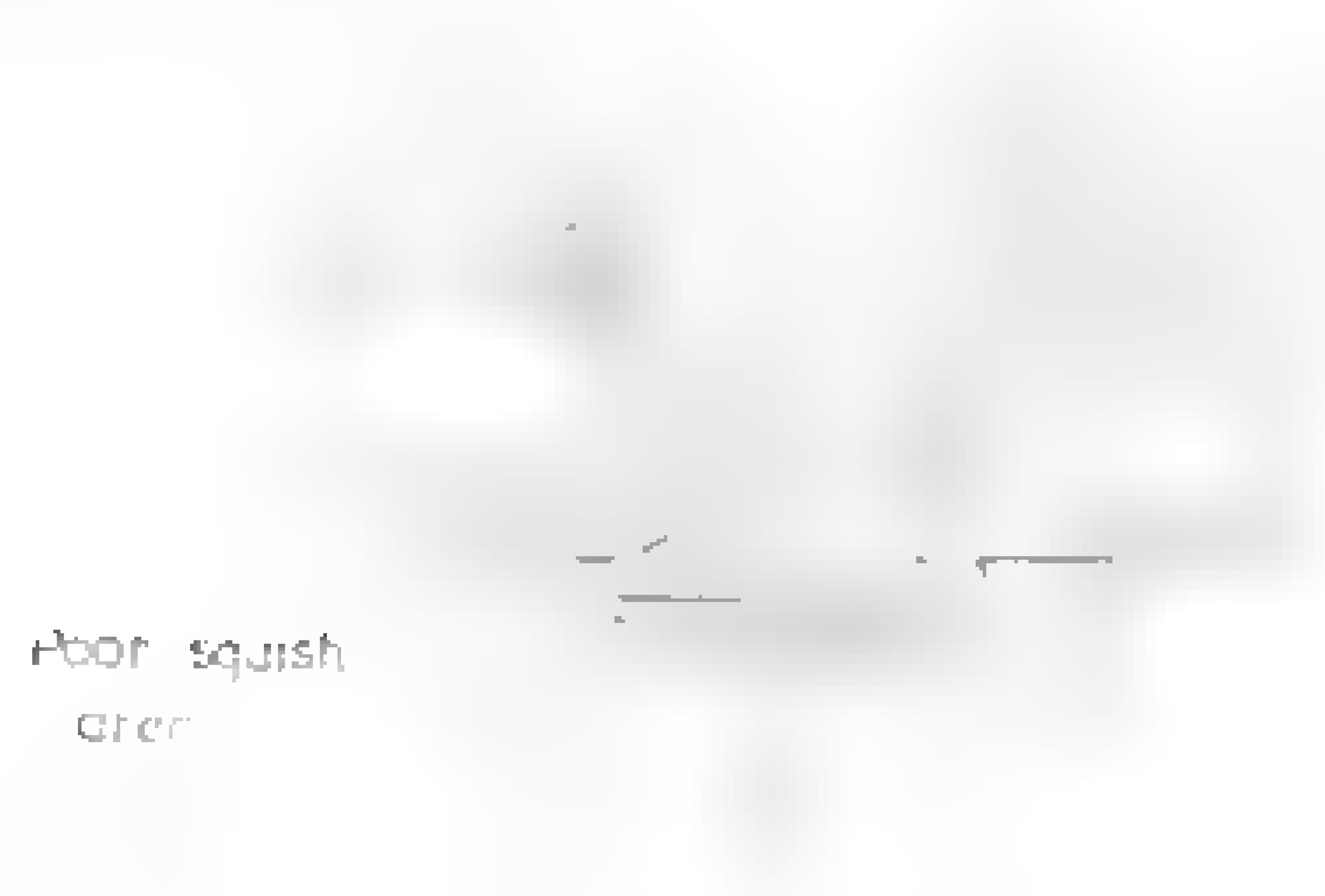


Figure 3.16 Downdraught conversion

into the cylinder in any given time. Figure 3.16 shows the Ford Anglia 105E head modified with downdraught inlet ports. On the 1,300cc Ford Escort, I have seen 143hp with this head on carburettors, which is all of 15hp, or 12%, more power than the same motor would produce on a 105E sidedraught head.

During the late 1970s many performance engines began to use what is known as the hemispherical chamber for this very reason. The inlet tract is straightened out and a large valve area is possible. This design, too, has had its problems. Back in the early 1950s the hemispherical chamber was being pioneered by such manufacturers as Jaguar, Chrysler and Norton. The design suffered a basic mixture burn problem, so 50° spark advance was commonplace. In Figure 3.17 you will note the valve angle (usually around 70–90° included angle), creating a huge combustion chamber. This large chamber surface area allows heat loss from the compressed mixture, reducing efficiency. Also, because of the large chamber area, the ignition flame must travel an extreme distance. Consequently two spark plugs were used in racing engines to aid combustion. Even today, after years of development, the Chrysler hemi and the air-cooled flat six Porsche still work better with two plugs.

Figure 3.17 Early Jaguar head design



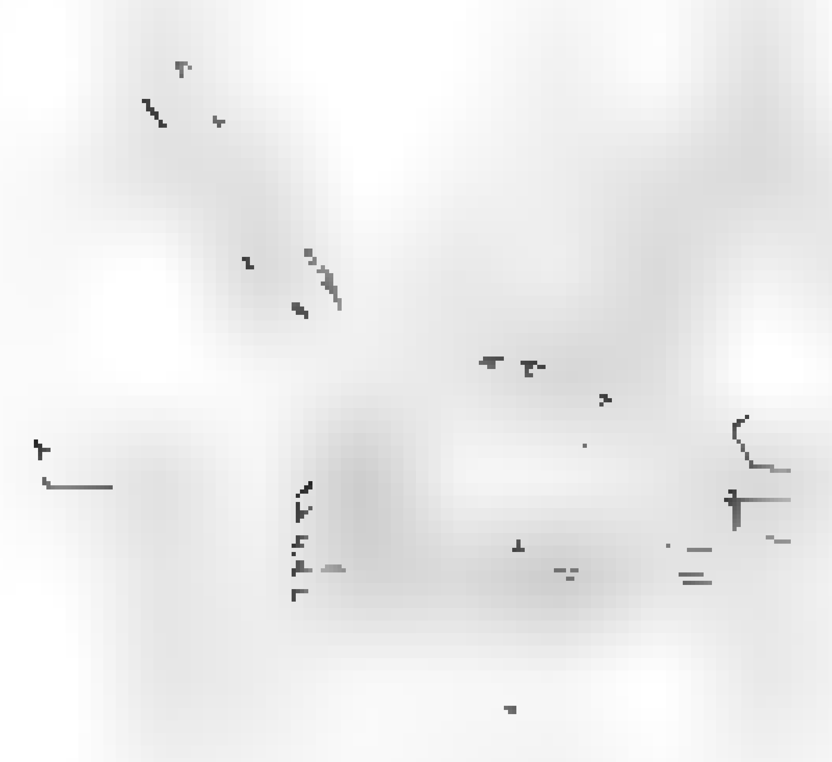
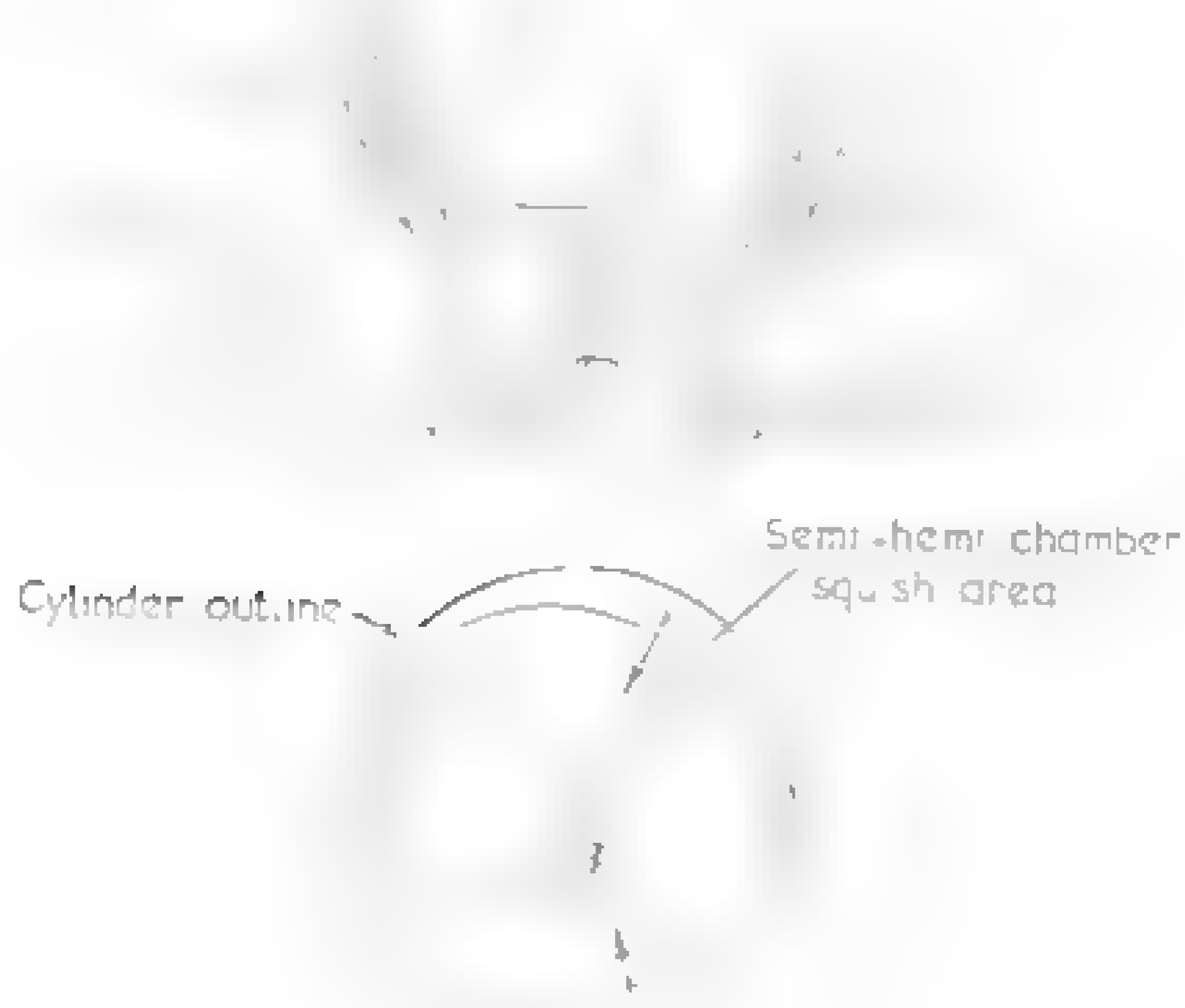


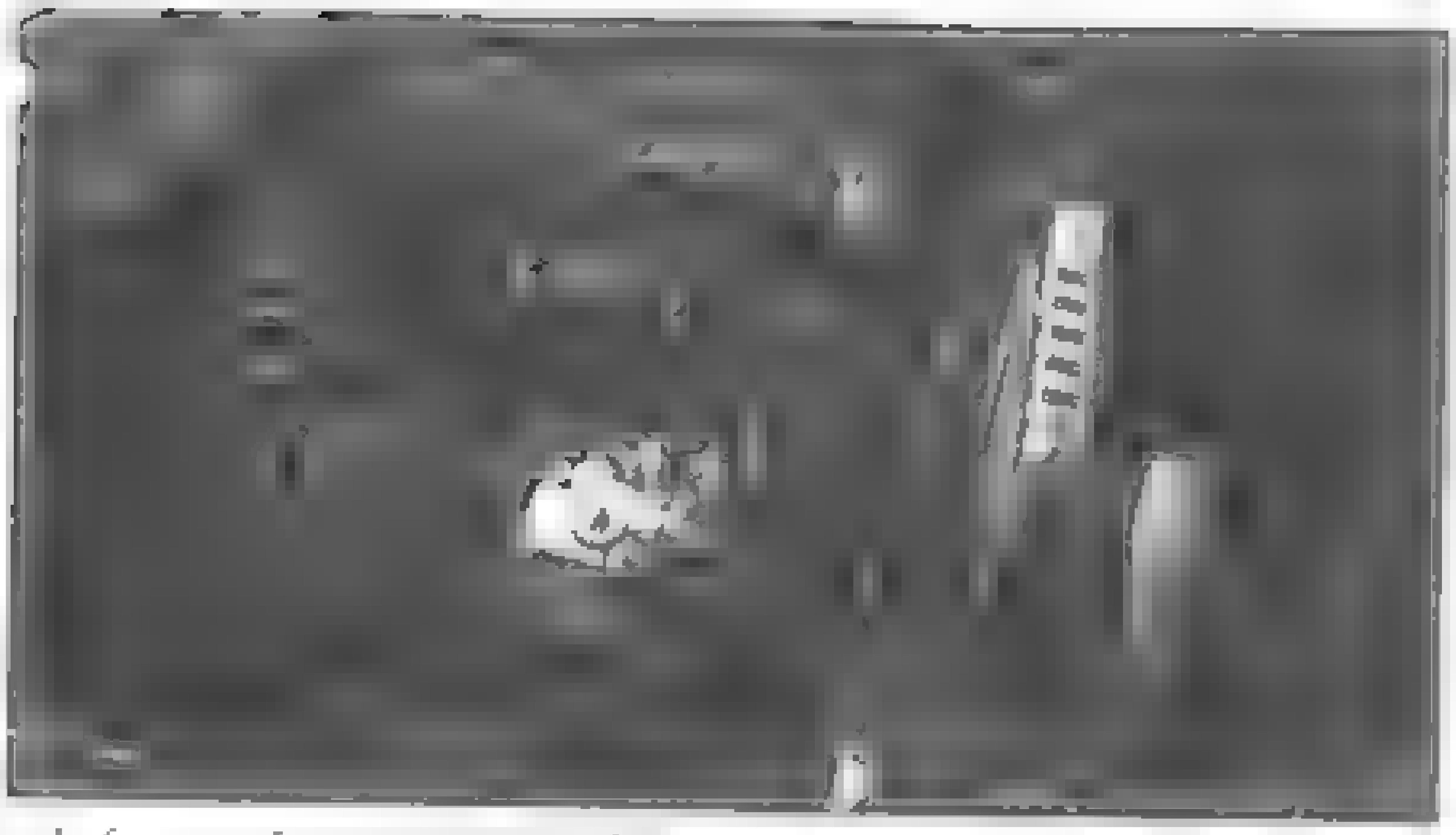
Figure 3.18 The later Chrysler Hemi has improved squish control

Remember our friend squish? Well, he is also lacking with this design. In Figure 3.17 you will notice that some squish does occur due to the radius of the [illegible] to promote good combustion

Later, the piston dome design was changed to that shown in Figure 3.18. This [illegible] is supplemented by a flat section running right round the crown, which comes into

Figure 3.19 Lotus/Ford Twin Cam head arrangement





...d purely from an hp perspective, the cylinder head Colin Chapman had Harry Mundy design for the Ford Cortina 116E engine was the best production 2-valve h... ever made. It was let down in competition b... ...row timing ch... water jacket t... t upset exhaust ...e cooling, both compromises being forced on Mun... ...need to make the engine fit ordinary product...

...contact with the lower reaches of the combustion chamber. Squash is improved bringing an improvement in combustion. This has resulted in higher power outputs and a reduction in spark advance.

In 1963 the brilliant Harry M... ...gned Lotus-Ford Twin C... introduced. Not a true hemi but a semi-he... this motor set the pace in head ... the car world at any rate, for man... ...ars. In 1.6-litre form this motor will produ... 210hp on carburettors, and will run quite happily with just a 30–34° spark lead. In fact the motor I ran on the road in my Ford Anglia for m...ny ye... ...ated with only 25 total advance. It was no slouch eith...r, producing 156hp at 6,800rpm.

The semi hemi co...bustion ...amber (Figure 3.19) has two distinct squ... ...t drive the fuel mixture towards t... spark plug. The valve angle has been c...sed up (54° on Lotus/Ford Twin C...m), allowing the ...e of a flat-top piston. Consequen... design enjoys a more compact combustion ch...mber than the true hemi... feature, combined with g... squish control, makes for high power outputs while sti... allowing a large valve area and a relatively straight inlet port.

You will also note that with this design, as with the hemi, the inlet valve... generally the exhaust valve, are com...tely unshrouded. This allows for fa... mixture flow right round the valve head. Instead of the valve flowing 80% ... and 20% on the other, as with the bath tub chamber, it will probably ... ...ething more like 57% and 43% in this instance. Therefore the available ... being used more effectively so that air flow and horsepower go up.

FOUR-VALVE PENT-ROOF CHAMBER

... This design ...
...by Ke...h Duck...

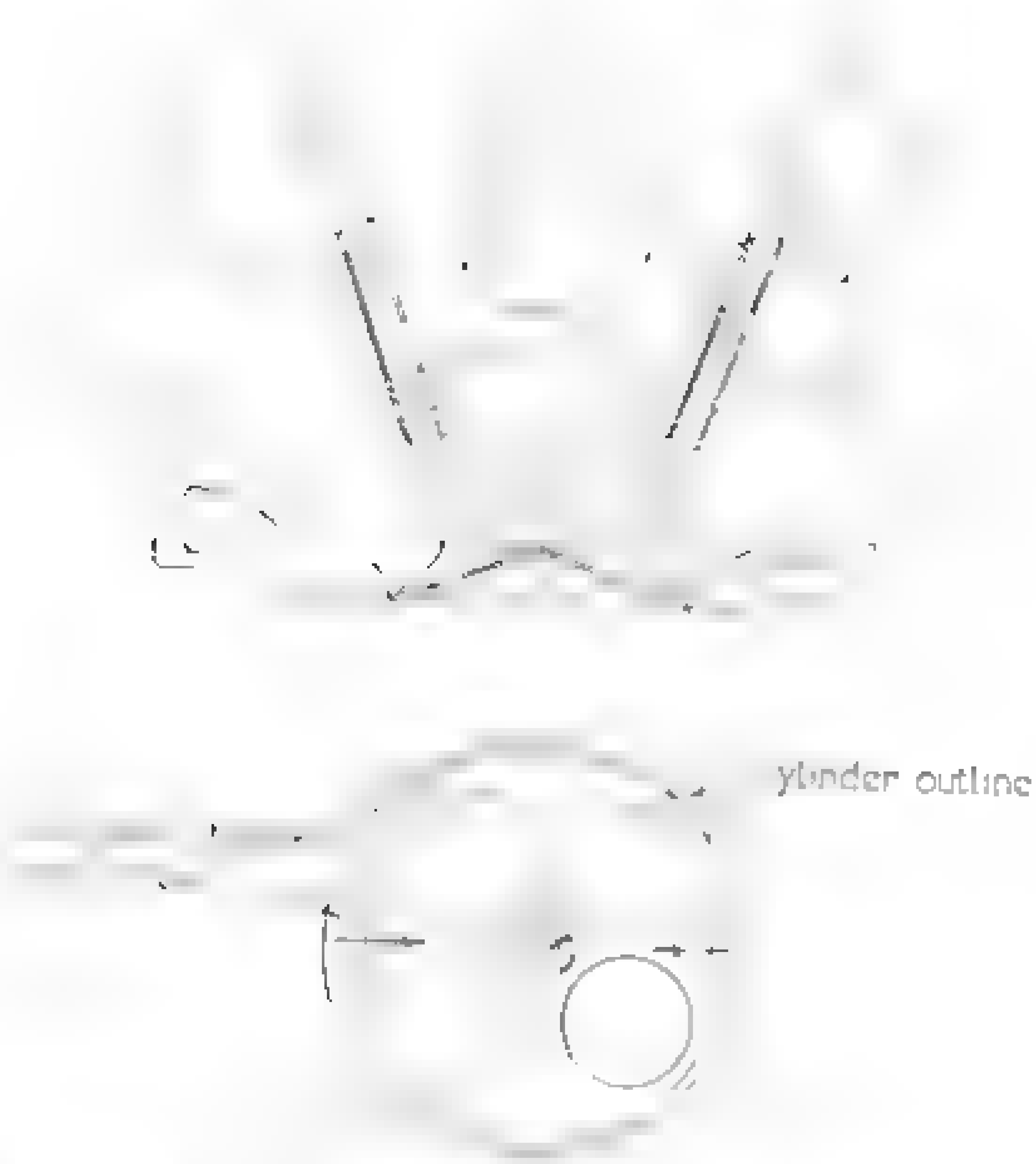


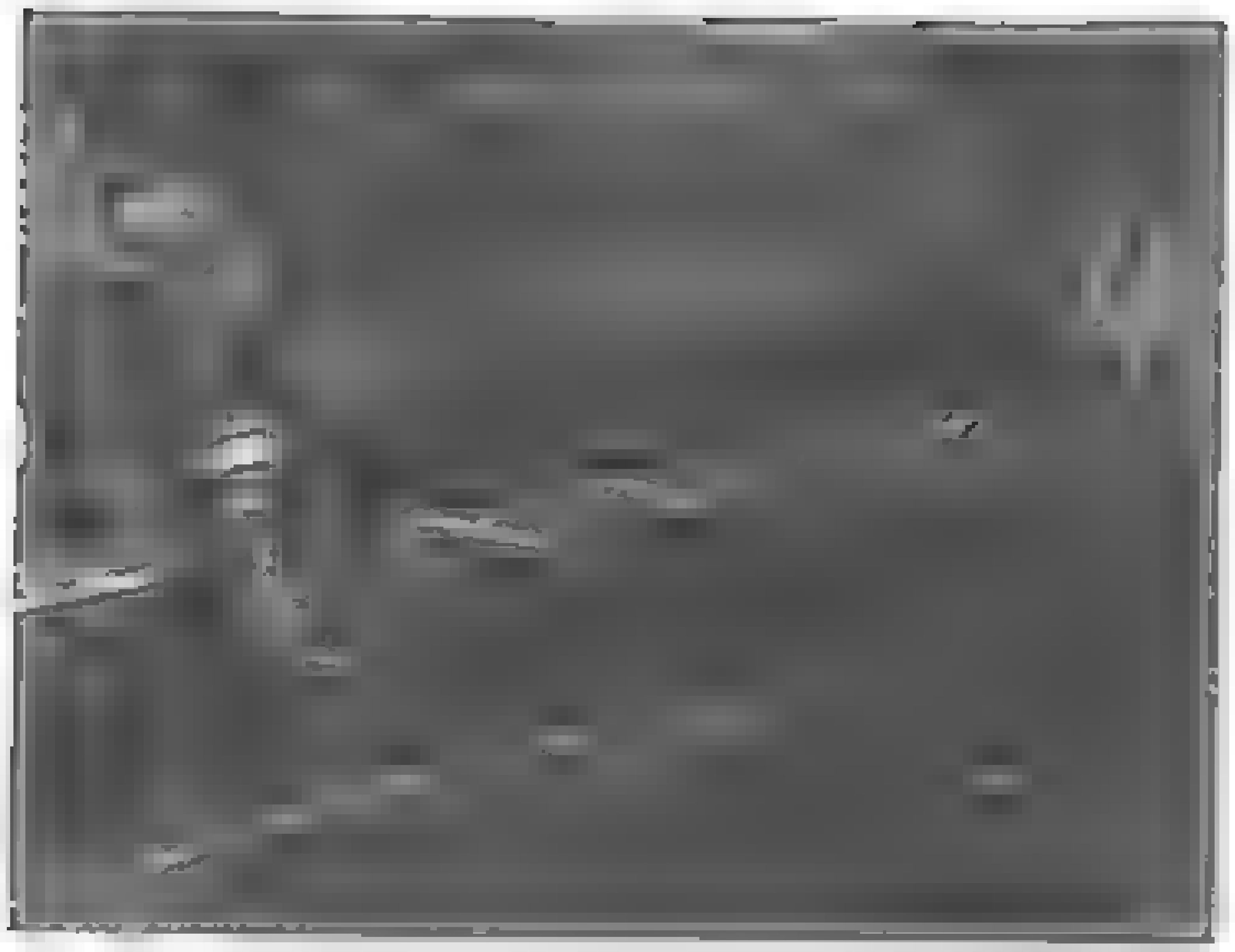
Figure 3.20 Cosworth Chevy Vega four-valve head

[illegible]
not so obvious.

Prior to World War I, Peugeot went racing with twin cam four-valve engines with success. Henri Peugeot's [illegible]
Mercedes, and later Maserati, at times also built four-valve race engines.

Both Coventry Climax and BRM also tried this layout in their Grand Prix [illegible]
equipped?

It was the motorcycle world that brought the four-valve design again to the top of race engine [illegible]

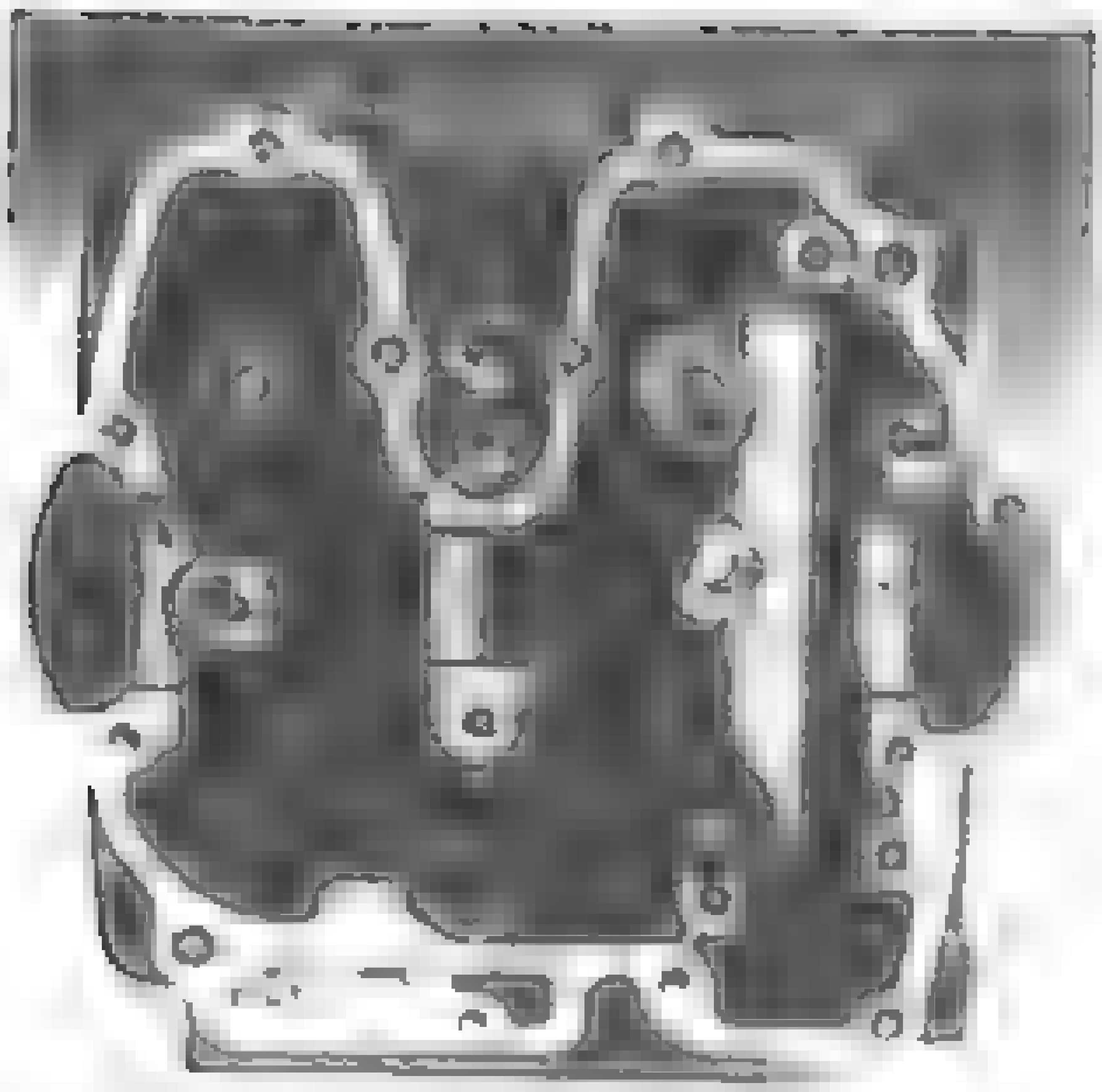


For 40 years the Duckworth designed 4-valve head for the Ford Cosworth DFV

chambers The 250 produced 60hp at 17,000rpm. Honda then carried the design the car Grand Prix circuit with their 160hp 1ltr Formula 2 engine. The two-va 140hp Cosworth SCA was no match, so when the Formula 2 capacity limit was lifted to 1.6 litres, Keith Duckworth designed his new motor, the Ford FVA, around four-valve pent-roof chamber. He got his design right, for the first time the FVA rode the dyno it pushed out 202hp at 9,000rpm; over 10% more than comparable two-va engines.

Initially Honda used a fairly wide valve angle of 63°, which required high-top pistons. Cosworth, however, closed the valve angle up to 40° (32° on DFV, DFX Indy turbo and stroked DFL sports racer), producing a very shallow combustion chamber which allows the use of flat-top pistons; ash is good and from opposite sides of a single, central, spark plug.

Now virtually every designer utilising the pent roof four-valve arr follows the design refined by Keith Duckworth. Over the years since le has been closed up even g valve area, to produce a flatter, mor combustion friendly chamber. sworth, now with Mario Illien heading design, produced a new Formula 1 engine dubbed the DFY. This had the inlet valve inclined at 10° and the exhausts at 12½° for an included angle of 22½°. Ferrari Tipo 049 Formula 1 eng 5°, but at the same ti big 6ltr V12 550 sports car Ferr chose 20°, 11° inlet and 9° exhaust. appears that around 20° includ now pretty much the accepted nor purpose designed race engines, but that obviously is not the case for other high pow applications. When designing the V12 6ltr Ast Martin DB9 engine, Coswort decided on a relatively wide 50° (25° for both inlet and exhaust), and the Porsche GT its valves at a more conventional 27.4° (14.7° inlet and 12.7° exhaust). Like th Porsche GT3 many sports bikes have an included angle around 2 comparably, usually a de on the inlet valv



Honda radial valve dirt bike head has valves tilted in two directions to move valve head away from cylinder wall and improve air flow

RADIAL OR COMPOUND FOUR-VALVE CHAMBER

In 1984, Honda released a number of single-cylinder dirt bikes in 250, 350 and 500 sizes with what they termed 'radial four-valve' cylinder heads, what today is more commonly called 'compound four-valve'. This development of the basic four-valve pent roof design involved moving the two inlet valves so that their valve stems were no longer parallel to each other, but were canted several degrees. The stems of the exhaust valves were similarly canted. As for the big block Mark IV Chev this arrangement permitted a small increase in valve size for any given cylinder bore diameter. More importantly, canting moves the opening valve heads away from the [illegible]

agitation and homogenisation.

Because the Honda was a single overhead cam engine they employed a complex [illegible] adopted this style of head an inverted bucket tappet, pinned to prevent rotation and [illegible] vertical there were cam lobe to cam follower friction and wear issues even with [illegible] there was freedom, from a mechanical standpoint, to go much more a[illegible] compression. Likewise when we apply deeper valve cut-outs to the piston crown necessary for the more steeply canted valve to avoid contact with the piston as it approaches and descends from top dead centre.

THE FOUR-VALVE ADVANTAGE

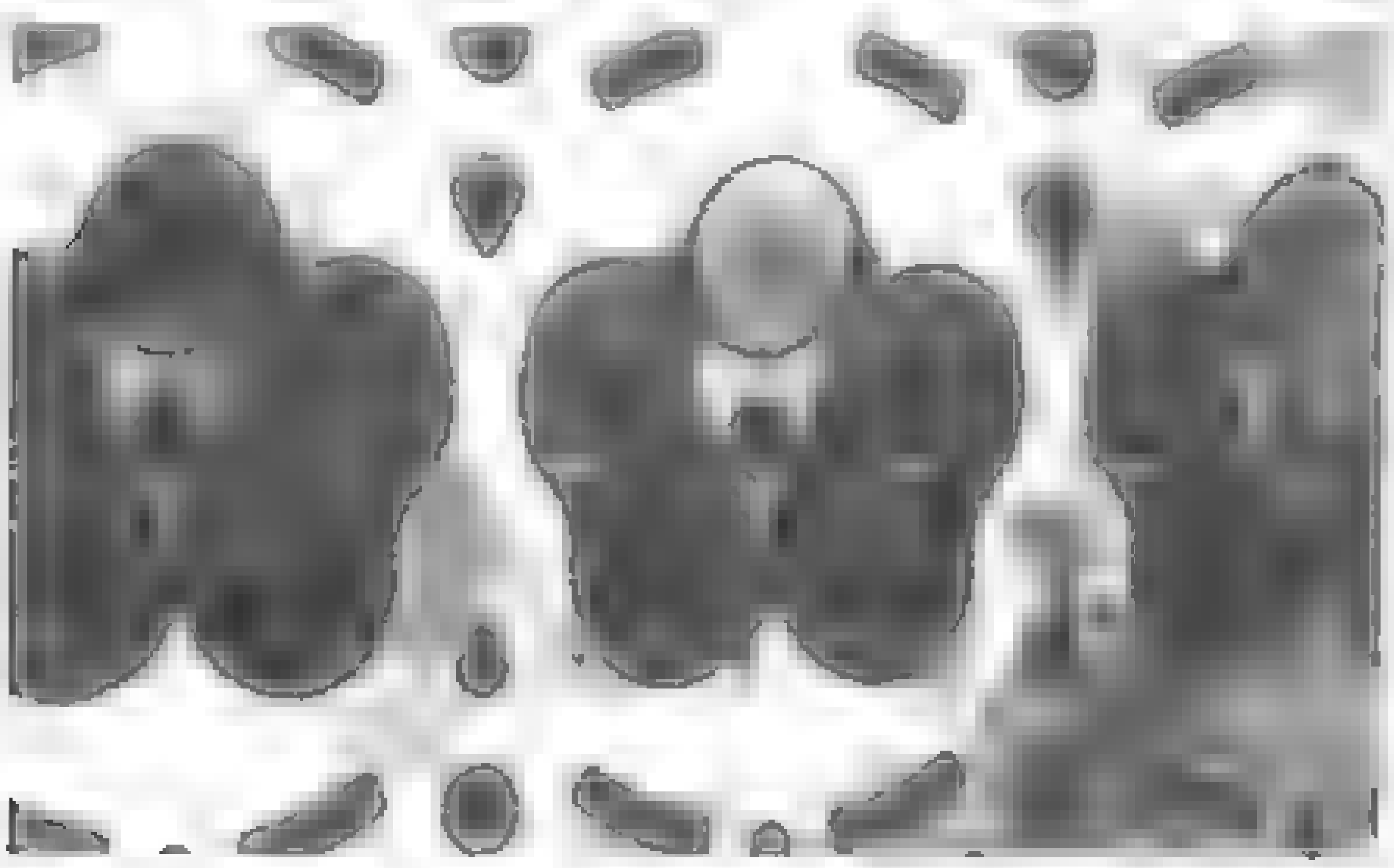
The arrangement of two inlet and two exhaust valves in the combustion chamber allows for an increase in valve area over that possible with a two-valve hemi, but contrary to what many believe, this is not the real reason for the superior performance of a four-valve engine. To give you an example of this, consider a 1,700cc Cosworth rally engine when compared to a 1,700cc Lotus/Ford Twin Cam rally engine. The Cosworth BDA has two 1.22in inlet valves, giving an area of 2.34sq in, while the Lotus/Ford Twin Cam has one 1.69in inlet valve with an almost identical valve area of 2.24sq in. Both engines develop maximum power at 8,000rpm; 190hp for the Cosworth, 170hp for the Lotus/Ford. The Cosworth is always much faster on any rally surface due to having a 1000rpm wider power band and far superior top-end and bottom-end power. This occurs because the Cosworth, although having a similar valve area, has 44% more valve flow area than the Lotus/Ford at all valve lifts. Remember the air does not flow through the valve but rather around the periphery of the valve, thus flow area is equal to valve circumference multiplied by valve lift. Therefore a four-valver can use short duration cams, which improve mid-range power without sacrificing maximum power.

THE FIVE-VALVE CHAMBER

A later development of the four-valve pent-roof chamber is the five-valve design brought to us by Yamaha. When the FZ 750 motorcycle appeared in 1985 it produced a true 86hp at 11,500rpm in standard road trim. Caring for the breathing of each of its 187cc cylinders were three tiny 0.827in inlet valves and two 0.906in exhaust valves. By comparison the then radical Suzuki GSX-R750 from the same era used two 0.984in inlets and two 0.945in exhaust valves.

In time a number of manufacturers tested the design but all struggled with excessive fuel consumption brought about by poor charge homogenisation and

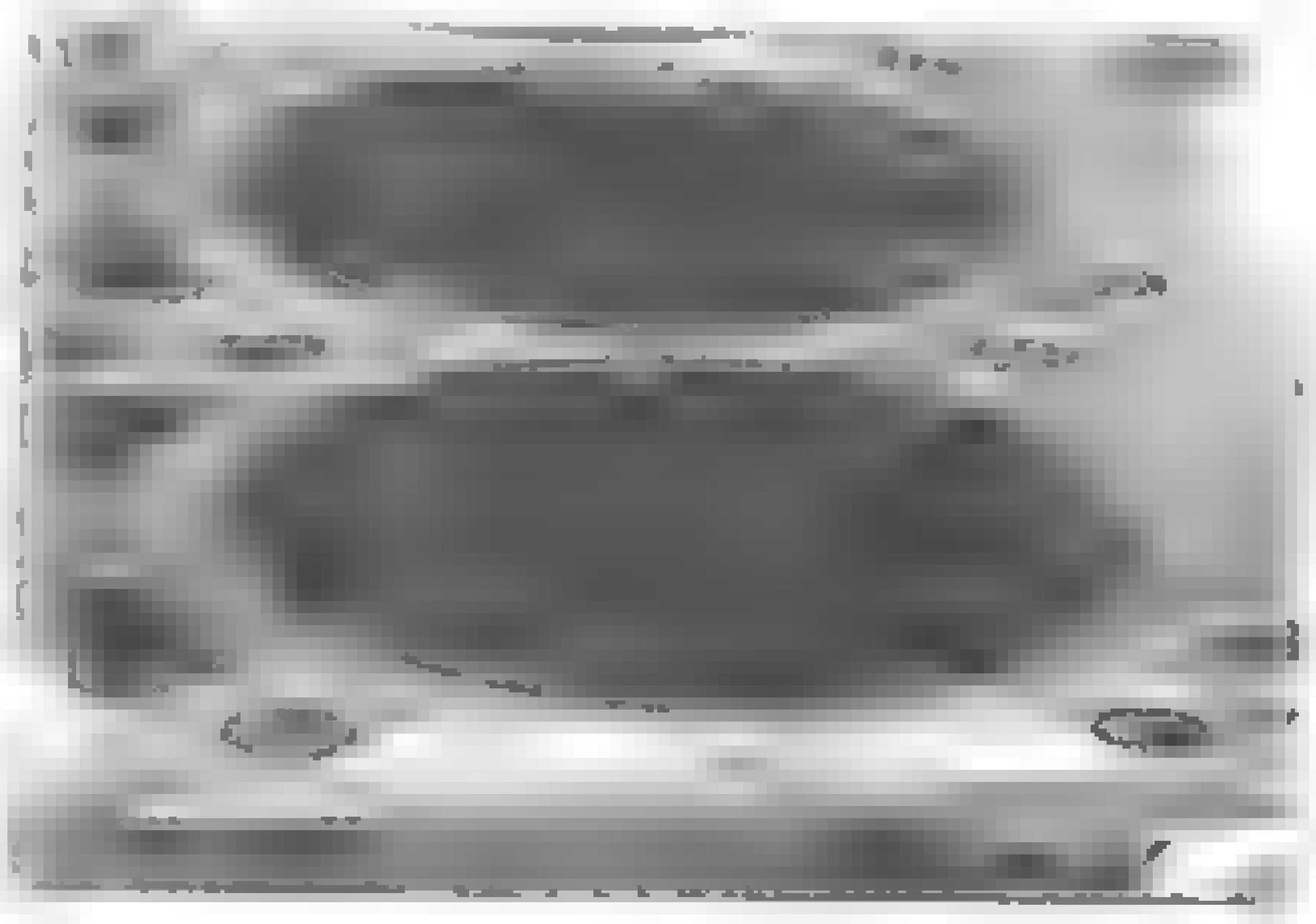
The Yamaha R7 5-valve chamber has good squish areas and inlet flow but the raised middle valve seat effectively divides the combustion chamber and hurts combustion.

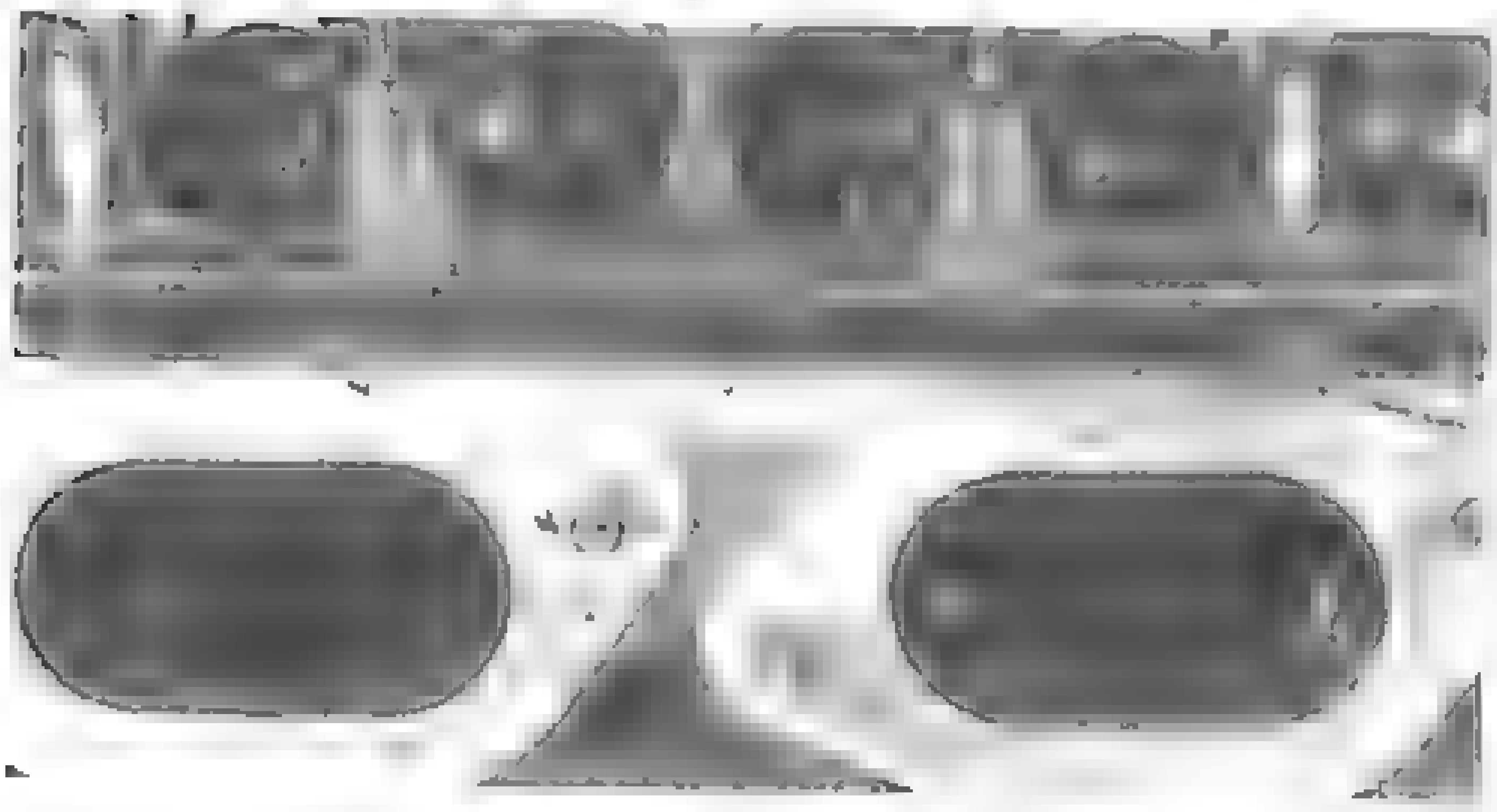


sluggish combustion. Sure the extra valve improves air flow into the engine, but a lot more fuel is needed to get power levels on a par with a four-valver. A couple of manufacturers [illegible] tool to give themselves a distinctive identity, but at this time even with the benefit of something like five redesigns by Yamaha, the five-valve head should be seen as a performance handicap relative to an average four-valve configuration.

The basic problems with this layout all stem from adding that fifth valve. First [illegible] is the lump of metal in the combustion chamber running from the seat of the extra [illegible] combustion space until the piston reverses, moving away from the head. Additionally like the dome on high compression pistons, the lump basically divides the combustion chamber into two. Thus when the piston rises on compression mixture homogenisation

[illegible] 5 valve Ferrari 360 chamber [illegible]rates limited squish and serious spark plug masking. In the inlet port the design doesn't provide good control of mixture flow direction as it enters the cylinder





...ing ... and fuel molecules not been ful... up and mi... cool pockets of mixt... allowed to form near the ... ater inlet valves. In turn, those two cool pockets ... is hurt relative to the amount of fuel consumed.

MODIFYING SEMI-HEMI AND FOUR-VALVE HEADS

The design of semi-hemi (eg Lotus Twin Cam) and four valve pent-roof heads ... usually quite good so not much can be done to easily improve upon th... ... the big hp increases that are possible with low tech push rod heads. The combustion chamber sometimes requires mild reshap... particularly if larger valves ... However, there are quite basic things you can do to improve flow in the inlet and exhaust ports. The main point to keep in mind is to keep your grinding tools away from the ports. You can clean out the 'dingle berries' but apart from that there isn't much else you should attempt without the aid of a flow bench. Really the part to focus your efforts on is the valve seat pr... and the area immediately under the valve seat in the throat area. Usually there are sharp ridges around where the valve seat meets the port that can be quite easily ground out. It is always difficult and expensiv... ...cable power increases from a head that is a good design to begin with.

Unfortunately, many motorcycle engines do not gain full benefit from incline... valves and hemi or pent-roof chambers, as the inlet port often has a sharp bend in it. On cars it is usually no big problem to fit the inlet manifold on downdraught or sem... ... that the carburettors or fuel injection clears the frame and fuel tank.

Main flow improvement comes from work in the inlet port valve bowl area. Valve seat insert overhanging the valve throat, the sharp lip where the machining meets the ... cast and port 'dingle berries' all upset fl...

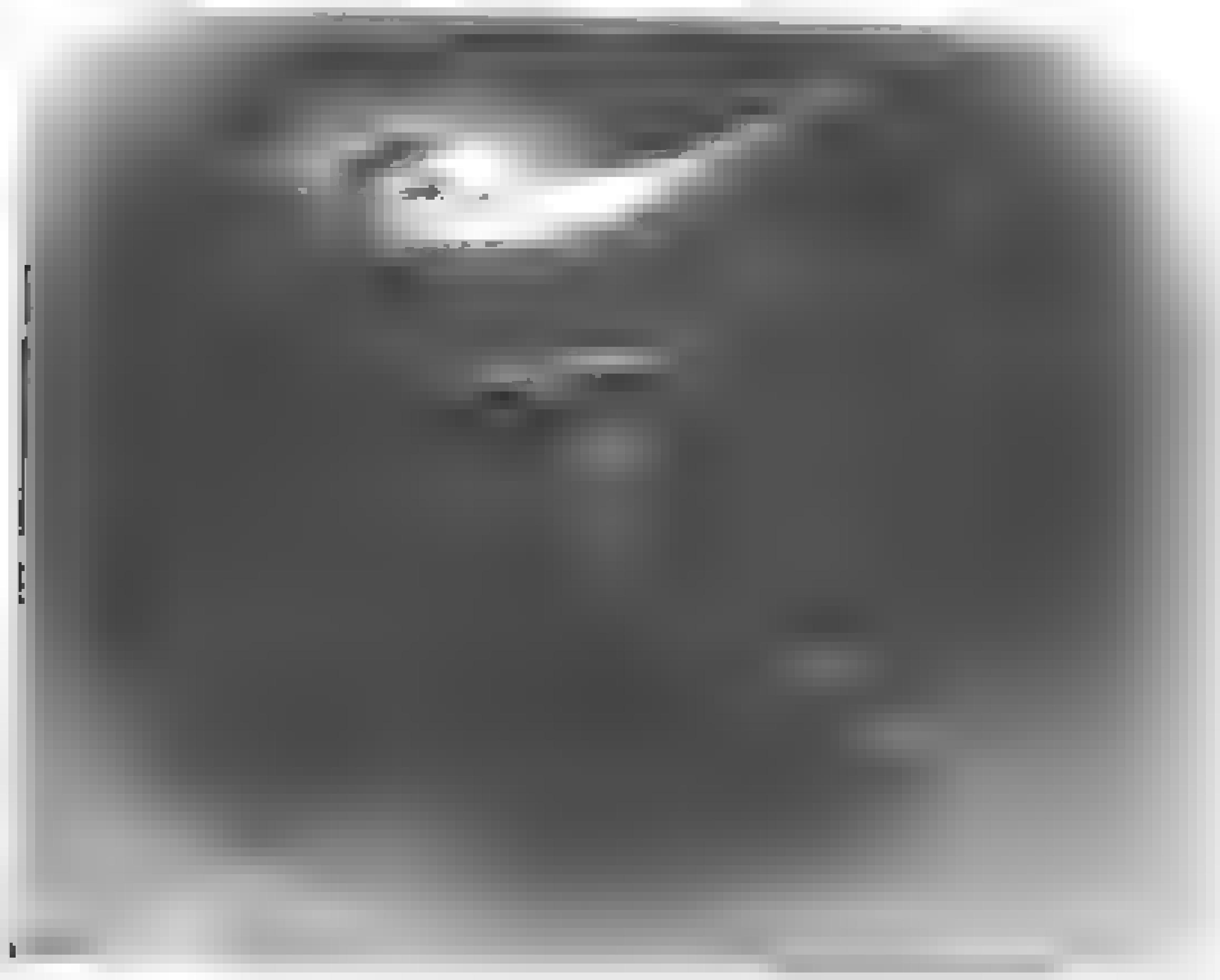


Figure 3.21 Motorcycle inlet port modification

This presents quite a problem as the inlet ports can usually be raised no more than about one-eighth of an inch, and this small amount does little to straighten out the bend into the combustion chamber. Usually all that can be done is to increase the radius of the bend. By grinding the top curve of the port, the top radius can be increased, but the amount of grinding is limited by the close proximity of the valve spring seat. The bottom side of the port is built up with epoxy or aluminium weld, and this effectively increases its radius. To further aid air flow, the port should be made gradually wider, so that at the bend it is no longer round, but oval.

A typical 1.2in round port would narrow down at the bend to a height of perhaps 0.9in (due to the small radius being welded up), and widen out to a width of around 1.45in (Figure 3.21).

VOLUMETRIC EFFICIENCY AND COMPRESSION RATIO

The next aspect to consider is the compression ratio, which is the relationship between the total volume of the cylinder, head gasket and combustion chamber, with the piston at bottom dead centre (BDC), and the volume contained in the space between the piston crown, head gasket and combustion chamber at top dead centre (TDC).

Changes in the compression ratio have a considerable effect on power output, because the higher the ratio, the higher the compression pressure at any given engine speed. As is true at all times, you cannot get something for nothing, which applies equally in this instance. An increase in compression ratio brings a corresponding increase in combustion temperature, so will the valves stand it? Bearing loads increase, as does the load on the ignition system, so do not rush in without considering the consequences. As a general rule, a road engine running on 95 to 98 octane petrol will be quite happy on a 9.5–10.5:1 ratio, while racing engines using 100 octane petrol usually run an 11–12.5:1 ratio and up to 13.5:1 on 100/130 Avgas fuel. With methanol, this can be increased to 14:1 or 15:1.

Above a true compression ratio of 14:1 no power is gained. However, if the engine has exceptional anti-detonation characteristics, a theoretical compression ratio of 15:1 may pick up a little power due to the fact that unsupercharged engines seldom, if ever, attain a volumetric efficiency of 100%. Simply stated, volumetric efficiency is the ability of an engine to fill its cylinders with mixture expressed as a percentage of the actual volume of the cylinders. Therefore a motor with cylinders of 100cc will be operating at 50% volumetric efficiency if it manages to get only 50cc of mixture into its cylinders. Obviously, such a motor, while operating on a theoretical compression ratio of 14.1, will, in fact, have a true compression ratio of only 7.1. As the average racing motor operates at approximately 93% volumetric efficiency, the theoretical compression ratio of 15:1 will actually be a true compression ratio of 14:1.

RELYING ON THE KNOCK SENSOR KILLS POWER

Some have got the mistaken idea that you can push the compression ratio way up, then just back off the spark advance, or let the knock sensor take care of it when the engine goes into detonation. The reasoning is that more compression means more hp, hence more must be better, and anyway isn't this what many car manufacturers do on their road cars? Don't they run the compression up high, then, when detonation is detected by the knock sensor, electronically retard the spark until knocking ceases? Yes, this is the way manufacturers design a number of their road cars, but remember they are chasing maximum performance, not necessarily maximum power. It all gets back to the volumetric efficiency. A road engine may spend much of its life cruising at 2,000–3,000rpm with the throttle just cracked open, so the cylinders are only being partially filled. Ideally, to obtain maximum cruise power and fuel economy the compression ratio needs to be pushed up, say, from 8.5.1 to 11.1, but here's the catch: all of these engines do not spend their entire time cruising. Hence when the accelerator is floored and more air gets into the cylinders, increasing the VE, the compression will now be too high and the engine will detonate.

Perhaps you are wondering why don't we have a variable compression engine that gives us an ideal compression ratio to suit various engine operating conditions. Yes, a bright idea, but we do not currently have those for passenger cars, so the manufacturers came up with a compromise. Instead of an 8.5.1 compression ratio that will not detonate on 92 octane fuel, they have pushed it up to say 9.3–9.8.1, not the best for cruise power and economy, but better than 8.5:1. However, this higher compression ratio will be destroying the engine by detonation at wide open throttle when the VE will be up around 87%. To avoid this a knock sensor is stuck on the

engine, and when internal hammering is detected the spark timing is backed off a few [illegible]. If the knocking does not stop the computer backs off the spark lead some more until it does.

However, this engine with a higher-than-ideal compression ratio running with [illegible] power, for improved part-throttle performance and reduced fuel consumption.

Performance and race engines spend much of their time not cruising at [illegible] the compression ratio is too high.

FIGURING THE IDEAL COMPRESSION RATIO

[illegible] breathes well at high rpm, filling its cylinders completely, will have to run a lower [illegible] the small block Chev V8 when modified for various types of competition. In 5-litre [illegible] engine, by virtue of its larger bore, can accommodate larger 2.2in inlet valves as [illegible] keep pace with the increased cylinder size, so the VE decreases.

Another aspect of how far we can go with compression ratios is vehicle weight [illegible] close ratio gearbox.

What then is the 'correct' compression ratio? Obviously this varies with engine [illegible] engine tuners spend a lot of time pondering this very problem.

On the dyno an 11.8:1 engine with 36° spark advance and an identical 12.3:1 engine with, say, 31° advance may give seemingly similar performances, but on the race circuit one may be superior to the other, or it may be superior on certain types of circuits (eg long, fast circuits). However, I have yet to see an engine give good hp that runs a compression ratio so high that it had to have the spark backed off more than a

The most suitable compression ra... ...t solely reliant on fuel octane ratings. W... Shell Racing 100 (MON 100) ... C-23 (MON 119) both permit very high ... ssion ratios, many ... rs also influence what is bes...

... ess than stock to k... ...t of detonation. Thus if the standard item the showroom wo... ed with 32° sp... ...e would expect a modified ... olerate 28° or more advance...

... nd.cator that th... ... good power with the camsh... ...ng in an advanced timin... ...rs to run with the cam in a split timing ... position. The net effect of this, along with ... ...ngine's power band and reduce ...

When calc... ... compression ratios a conservative rule of thum... ...ure of 0.115 multiplied by the Motor Octane Number (MON) of the ... ...n that particular class of competition. Thus, if you are running Av... ...lly 100 MON), you would start out with a compression ratio of 11.5 ... ...ld be a good starting point for an engine with carburettors requir... ...road power band. To begin our testing on electronic fuel injection we would ... 0.118 multiplied by the MON. At the other end of the scale, an engi... ...ry soph... ...d electronic engine management, operating on a relatively ...

... be o... ...
...1 MON f...

MEASURING THE COMPRESSION RATIO

It should be noted at this stage that quoted compression ratio figures are always theoretical compression ratios and are expressed by the following formula

$$CR = \frac{CV + CCV}{CCV}$$

CV is the cylinder volume, and is easily found by dividing the engine capacity by the number of cylinders. Therefore a 2,000cc four-cylinder motor has a cylinder volume (or swept volume) of 500cc
CCV is the combustion chamber volume, and is not so easy to calculate. It is made up of the combustion chamber volume, plus the volume that remains above the piston when the piston is at TDC, plus the volume caused by the thickness of the head gasket, plus the volume of the dish if dished pistons are used, or minus the amount displaced if high-top pistons are used.

Assuming that the pistons are flat tops, the formula for finding the volume of the head gasket or the volume above the piston is:

$$V = \frac{\pi D^2 \times H}{4000}$$

where $\pi = 3.1416$, D = the diameter of the bore in mm, and H = the compressed thickness of the head gasket or the clearance between the piston crown and the block deck in mm

The volume of the combustion chamber is measured as shown in Figure 3.22, using a burette filled with liquid paraffin. If the head has been modified by a tuner, he should

Figure 3.22 Measuring combustion chamber volume.

formula

$$V = \frac{\pi D^2 \times H}{4000}$$

this volume should be

$$V = \frac{\pi \times 102^2 \times 13}{4000}$$

which is 106 2cc

However, measuring the volume with a burette we find it to be only 74 7cc Therefore

volumes we must go back to the original formula to calculate the compression ratio

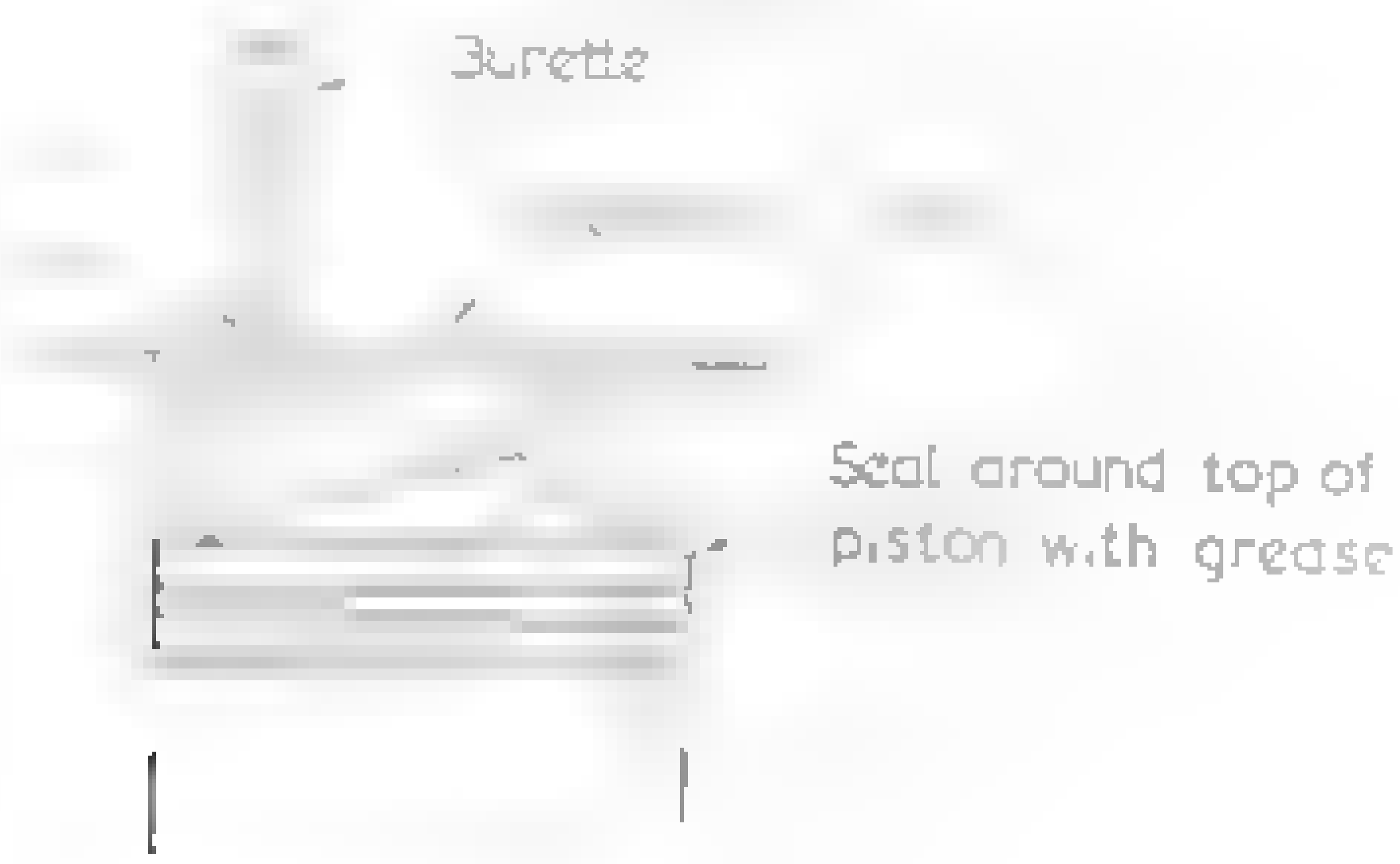
VALVE MATERIALS AND CONSTRUCTION

Today's engines make extremely taxing demands on the valves. Let us just look at a

against the valve seat and blasted with a hot flame that will raise its

pressure of around 1,500 pounds per square inch (psi)

Figure 3.23 Measuring high-top piston displacement



To exist under such conditions, a valve has to be made of pretty good material, but here we have a problem. Racing valves are made of austenitic stainless steel which, while being able to withstand high combustion temperatures, has very poor scuff resistance. Therefore they should be used only with valve guides made of silicon aluminium bronze, which is more compatible with austenitic steel as far as anti-scuffing is concerned, and of course bronze aids with heat transfer. The type of material we want to use in the valves is something like KE965, EN54, 21/4N or Nimonic 80a steel.

Most current production valves are of bi-metal or tri-metal construction, ie the valve head and stem are of different materials, welded together, in order to overcome the scuffing problem mentioned earlier. The result is that we have a head made of quality austenitic steel such as 21/4N and a stem made of something like EN8, which is perfectly happy running in a cast iron valve guide. A valve made of such material is suitable for a high-performance or budget racing engine.

The way to test if the valve is of austenitic steel is to see if it is magnetic. This applies to all valves: if they are magnetic they are no good. If the stem is magnetic and the head is non-magnetic, it is a welded valve, which is all right. Check before you buy any worked over heads or valves to see that you are getting good examples. Generally, production inlet valves use a head of EN52 steel welded to a stem of EN8, which is not quite good enough, unless you don't plan on more than a modest power increase; this type of valve has a magnetic head and stem.

If the budget and race regs allow them then we want to do all we can to reduce valve weight by using either hollow stem or titanium valves. Titanium is preferable, dropping valve weight by about 35%. However, if the rules insist on stainless steel, going the hollow stem route may be a viable alternative. This can save up to 20% weight in valves with thicker stems over about 8mm.

In addition to the valve's mechanical durability we have to do everything possible to preserve the integrity of the valve seat. As the seat wears power drops off due to a poorer gas seal and poorer air flow over the resculpted seat surface. Typically this amounts to around a 1% hp reduction after 1,000km. And because of that wear expensive valves that may be mechanically sound will be tossed so as to maintain performance. To get longer valve life and keep the seat in good shape, application of a DLC makes good sense. Anatech's Casidiam coating is an extremely hard and flexible carbon coating that won't chip and will cut seat wear to a quarter of that of an uncoated valve. Applied to steel it cuts the coefficient of friction by 45% and on titanium it's down by 75%.

VALVE GUIDE MATERIAL AND CLEARANCE

A little earlier we mentioned the valve guide's role of transferring heat, from the exhaust valve in particular, to the engine coolant. An equally important aspect of the valve guide is its role with reference to engine performance. As its name implies, the guide is responsible for guiding the valve accurately on to the seat, to effect a gas seal. A worn guide cannot do this, so power is lost. If the inlet guide is worn, it will allow oil into the engine, upsetting combustion and perhaps cause detonation.

Just as too much clearance between the valve stem and guide is to be avoided, so is too little clearance. In this situation the exhaust valve can seize in the guide and

Sloppy valve guides add up to an overheated exhaust valve. Once the hot gases begin eroding the seat area this soon happens

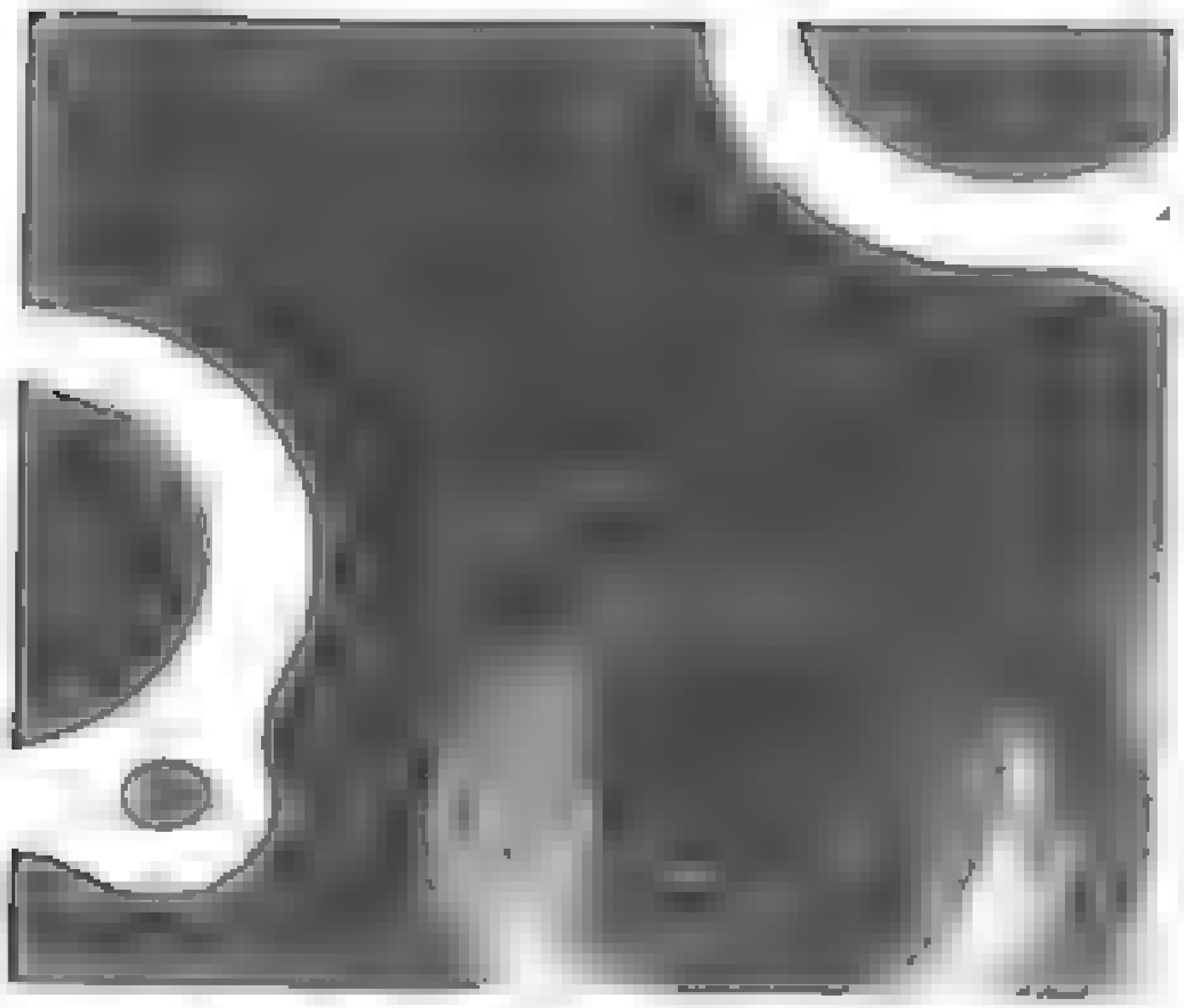
knock a hole in the piston crown. Air-cooled and supercharged engines are prone to this problem, so be sure to measure the clearance – do not just rely on your sense of feel. Generally, the clearance should be from 0.0015–0.003in for inlet valves and 0.002–0.0045in for exhaust valves.

However, naturally aspirated water-cooled engines with high-oil-retention valve guide materials like silicon aluminium bronze and silicon copper nickel (Colsibro) should run much tighter clearances, 0.0007–0.0012in for the inlets and 0.0012–0.0017in for the exhaust valves.

VALVE STEM SEALS

[illegible] used will depend on how much space there is for a seal between the valve stem and [illegible] valve heads as possible. Carbon deposits not only cut gas flow into and out of the [illegible] the exhaust valve, wh[illegible] is operating at very high temperature.

Neoprene seals as fitted by many car manufacturers can be suitable but are not [illegible] with the resulting problem of carbon on the valve head and exhaust smoke on startin up the engine, or too dry, which leads to premature stem/guide wear and possible seizure. Fitted to the exhausts they do not seem to handle the heat too well, so they quickly harden and fail to provide good oil control.



It is important to ensure that there is sufficient clearance for valve stem oil seals between the valve cap and valve guide at maximum [illegible]

[illegible]rfect Circle-type teflon seals are preferred for competition engines. Like [illegible]e seals they slip over the top of the valve guide, but in race engines that [illegible] [illegible]um valve-stem-to-guide clearance, and hence no valve stem wobble, they do a more consistent job of controlling valve stem oil flow. However, when the clearance increases they work quite poorly, so if you allow your engines to run down like this you may be better off with another type of seal. When fitting these seals over exhaust valve guides it is a good idea to lightly ream them, using a reamer the same size as the valve stem diameter to allow a little more oil flow to the guide.

A less hi-tech solution is to fit an 'O' ring seal up under the valve collets (valve locks) and an oil splash shield under the spring retainer. The 'O' ring prevents much of the oil flowing from the camshaft and/or rocker arms from travelling down the valve

Oil splash [illegible]lds can be fitted under the valve caps to direct oil away from the valve stem [illegible] *ving through the collet*

stem in the gaps between the collets. Instead, most of the oil has to flow out over the valve cap and the oil splash shield fitted under it, then down over the valve spring. This doesn't work on the inlet valve as well as a neoprene or Perfect Circle teflon seal [illegible] away heat, valve springs begin to fatigue and lose pressure.

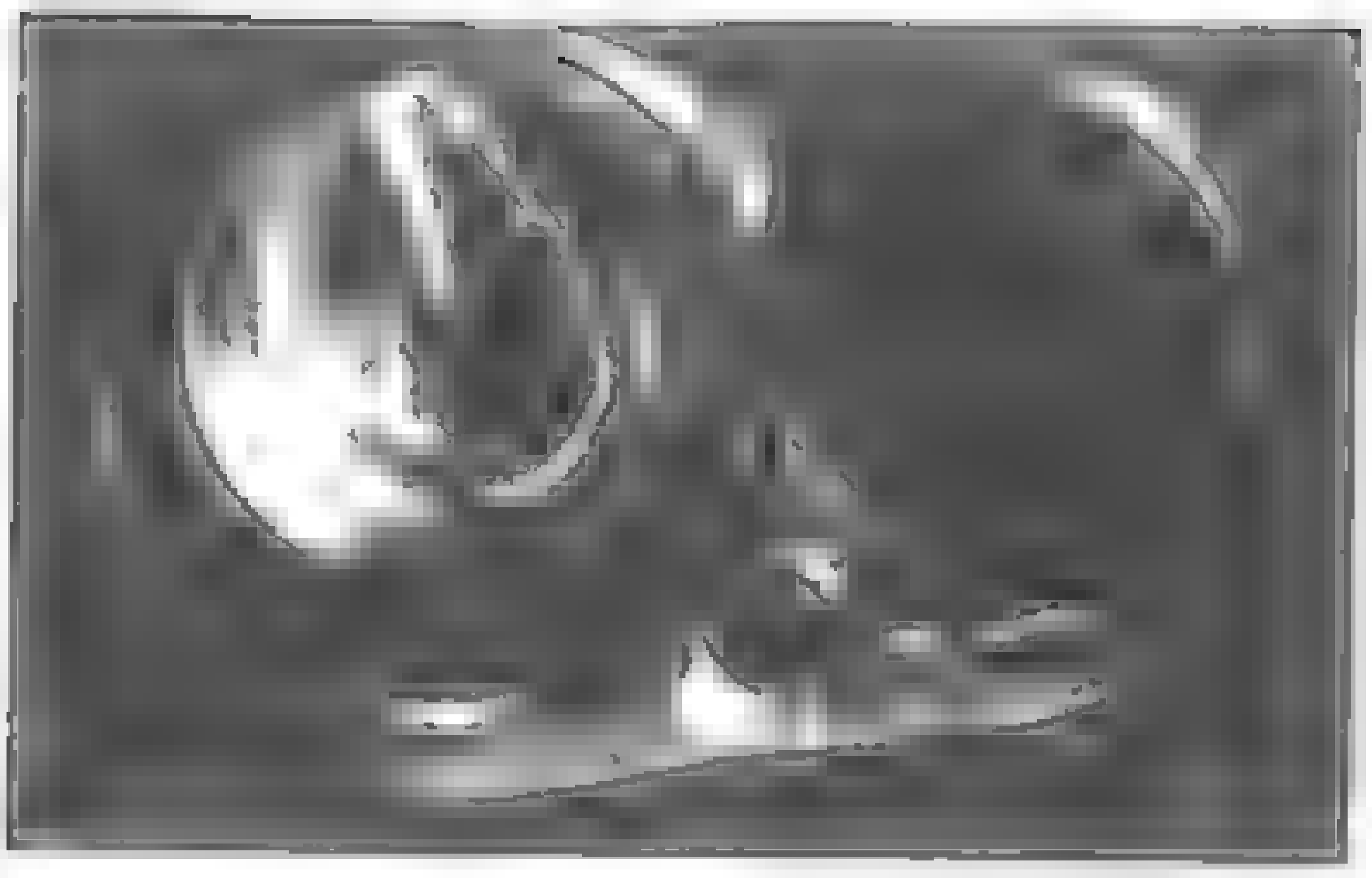
Another solution that works in the same manner as an 'O' ring, but more effectively, [illegible] assembled. Note that for this to work properly the top of the valve stem, the collets and [illegible] type engines.

In engines with large-diameter inner springs, the old 'umbrella'-type seal can [illegible] rectified, then any appropriate valve stem oil control system can be chosen.

HOW TO CHOOSE WISELY

Armed with all this information on cylinder heads you are in a good position to make a [illegible] must be kept in mind particularly if you are running a competition motor where [illegible] remains clinging to the port, causing a flat spot and power loss.

Also don't be fooled into thinking that a CNC ported head is better than a hand [illegible] proved to work was finally settled upon was the expensive task undertaken of [illegible] great. If the porter knew his stuff and the hand ported head worked very well, then the CNC clone will be just as good, and it will be consistently to the same standard in every

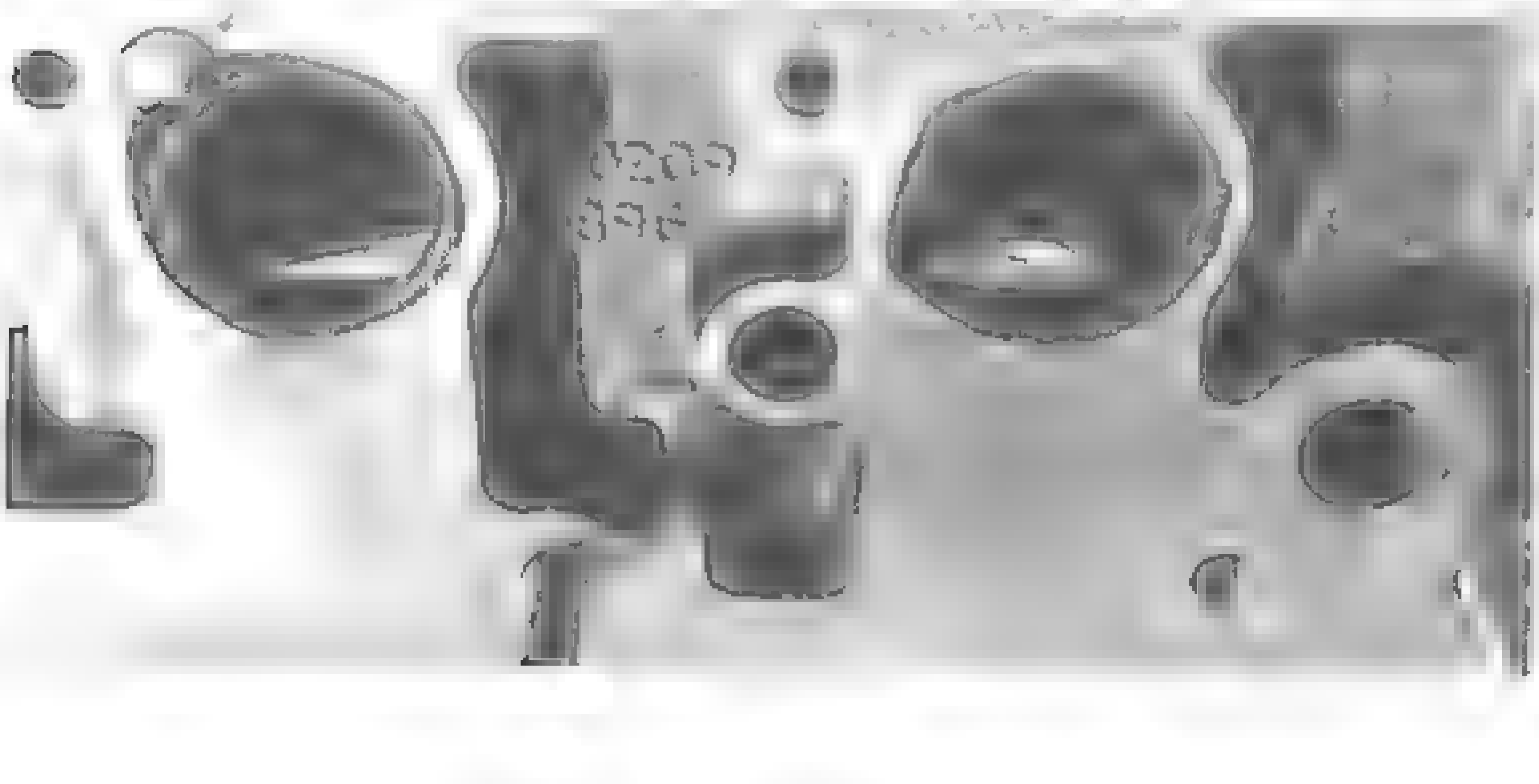


The tool marks in the valve bowl area indicate that this is a CNC ported head. How well it performs is dependent on many factors.

port and every combustion chamber. The CNC machine doesn't produce any Friday or Monday heads. It doesn't make any major or minor slip-ups. However, if the original hand ported head was a dud or not suited to your application then the CNC clone of that head can only deliver the very same unsatisfactory result. The situation is the same if the CNC head wasn't a true clone of the good hand ported original, it simply won't work as well as the original. The latter occurs for a number of reasons: perhaps the CNC programming is poor or incorrect, maybe corners have been cut to reduce the number and/or complexity of machining operations, or it could simply be that an old worn out machine or a new less sophisticated machine is performing the work.

Before you take delivery of the ported head it should be matched to the inlet manifold. At the same time studs or dowels should be fitted so that each time the manifold is removed, it will, on being replaced, align perfectly with the ports in the head. These dowels need only be one-eighth of an inch diameter silver steel.

The inlet ports are scribed to match the inlet manifold. As well as machining to eliminate the lip either dowels or studs are required to ensure the ports always align.



While the head porting work is being done, other specialised work should also be carried out such as re cutting the valve spring pads. Therefore the head modifier will have to be given specific instructions by you regarding such things as what valve

what rocker stud modifications are required

CYLINDER BLOCK PREPARATION

At the same time you should find out if you need to do any grinding of the cylinder

going down the side of the piston

The deck of the block needs to be perfectly flat and completely free of any foreign

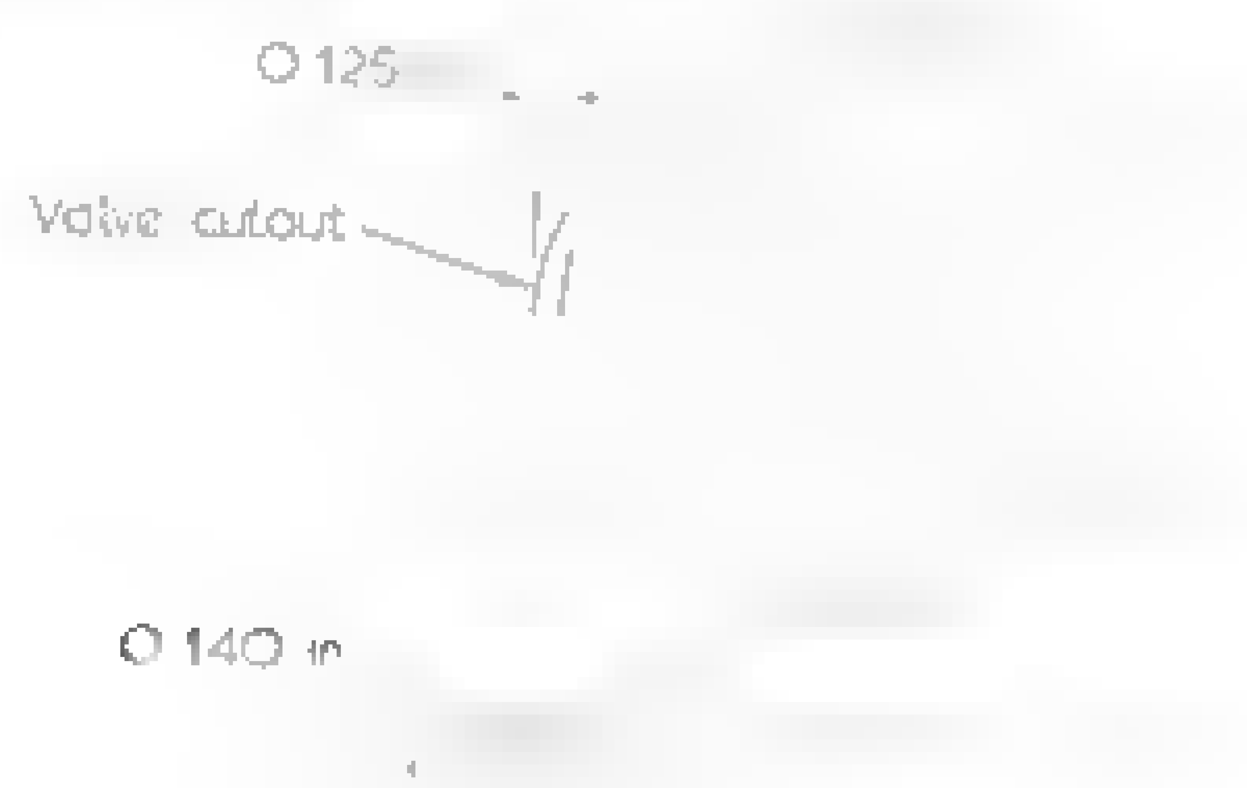
scraper. If the block has dowels or studs still in place be extremely careful to cut in

paper or wet-and-dry paper as abrasive particles will undoubtedly end up in the cylinders, and in push-rod engines you can add the cam and lifters too. Don't think that compressed air will help – most likely it will spread the grit deeper into the engine

HEAD GASKET SELECTION

Obviously all this preparation is intended to provide a nice flat bed for the head and

Figure 3.24 Austin Mini exhaust valve cut-out



However, it isn't high cylinder pressures as such that gives head gaskets a hard time. The real culprit is detonation, so if we keep the engine free of detonation blown head gaskets generally will not be a problem.

Copper asbestos gaskets are definitely out as the thin annealed copper sheet is just too soft to hold up. On the other hand, stainless steel sheet gaskets are too hard to deform into minute surface imperfections, so give trouble too. Steel/copper/asbestos head [illegible] and cylinder fire ring, with copper sheet on the other face, and asbestos in between. Some high performance composite gaskets use a special cylinder fire ring which is better [illegible]

[illegible] 'O' ring, but the mild steel round wire 'O' ring may also be suitable. However, the latter is more likely to be used with a solid copper type gasket. The usual procedure with steel 'O' rings is to machine the block to accept 0.041in mild steel wire, such that it stands proud of the deck about 0.006–0.008in. As such it exerts a considerable compressive force against the gasket and head.

HEAD GASKET CLAMPING FORCE

Apart from detonation the other big destroyer of head gaskets is insufficient clamping force between the head and block, allowing the gasket fire ring to flap about in response to the violent fire storm going on in the combustion chamber. We have already discussed the need for a perfectly clean and flat deck surface on both the head and block, and later we will cover correct head tension and tensioning technique. That [illegible] and/or head and also insufficient and/or inefficient clamping devices – read head studs or bolts – around each cylinder bore.

The most common cure for blown head gaskets is to fit head studs if the manufacturer is relying on bolts to clamp the head to the block. Studs clamp better than bolts and they cause less distortion and pulling of the bock deck (which is none too solid in some engines), so that sealing is improved. When initially fitting the studs, torque them to 30lb ft. Remember that when studs are fitted to Vee-style engines it will often be impossible to remove the heads, due to a lack of clearance, unless the engine is first lifted out of the vehicle.

With some engines the factory has deleted bolts where they were too difficult to fit in a normal assembly line situation. However, for a high performance or race engine we may be able to devise a method of adding those missing bolts and it is amazing just how four additional $^{5}/_{16}$in NC bolts tensioned to 18–20ft lb cures the sealing problems of powerful big block Chevys.

After this we have to consider the integrity of the deck of the cylinder head and

[illegible]

HEAD BOLTS AND TENSIONING

[illegible]

(alloy or cast iron) should be tensioned again

One thing to keep in mind when tensioning head studs or bolts on an engine that

[illegible]

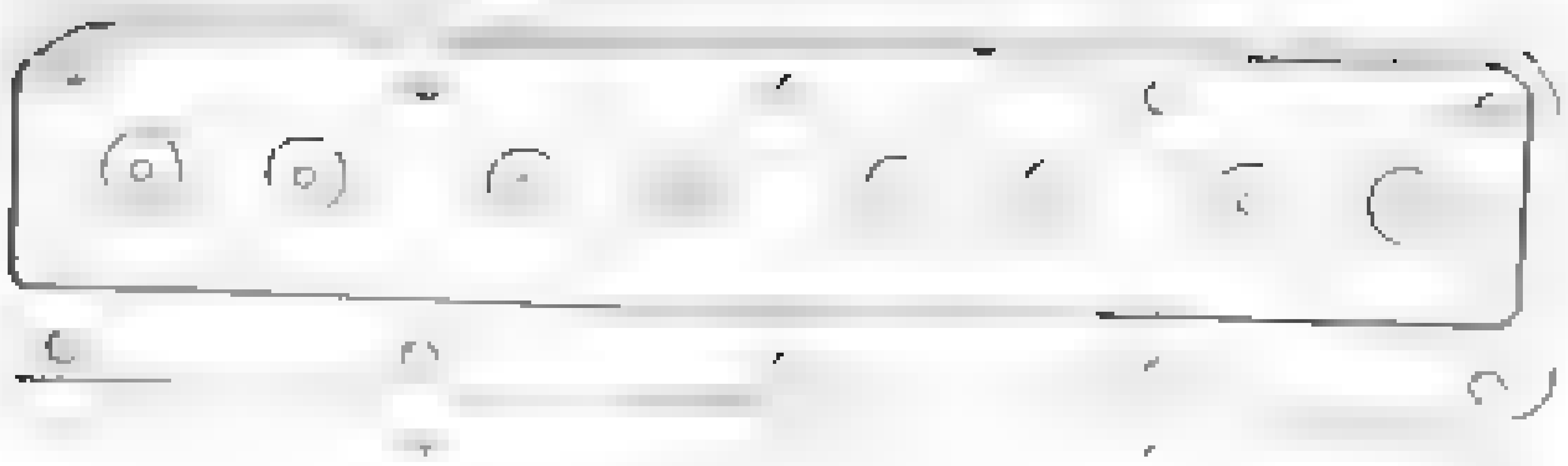


Figure 3.25 Cylinder head bolt tightening sequence

[illegible] is 65lbf ft, torque the short studs to this figure and the long studs to 72lbf ft

CYLINDER HEAD HEAT TREATMENT FAILURE

[illegible]

[illegible] couple of places. A casting flaw could throw the reading way off, so over an area of about a square inch test in at least 10–12 spots.

If the head is soft and you have a lot of money invested in it, it may be [illegible] in heads where the cam runs directly on the aluminium. Ask around to find what [illegible]

[illegible] 20 hours), then allowed to cool slowly in the air. Following this the head will be machined for oversize guides and seats and the deck will have to be resurfaced. This all costs a lot, but if the head is a good one with lots of fancy port and chamber work you will not want to throw it away. Note also that any time a head is welded close to a head gasket fire ring line or valve seat it will also require a fresh heat treat [illegible]

Chapter 4

Carburation

T[illegible] performance modifications should begin. A certain amount of glamour and fascination has always surrounded large or multiple carburettors, and as a result unwary enthusiasts can often be caught out with mixing devices that are either too [illegible] and excessive fuel consumption.

The basic requirement of a performance carburettor is that it mixes the fuel and [illegible] reason we have to be very selective as to the type and size of carburettor that we choose for our particular engine.

Table 4.1 High-performance engine fuel–air requirements

Running condition	**Mixing ratio (by weight)** *Fuel : air*
Starting	[illegible]
Idling	[illegible]
Low-speed running	[illegible]
Light-load ordinary running	[illegible]
Heavy-load running	[illegible]

To understand more fully what we should be looking for in a high performance carburettor, we need to go back to the basics and get to know how a carburettor works.

system, and also an acceleration pump system and a power system

FUEL INLET SYSTEM

The inlet system consists of the fuel bowl, the float, and the needle and seat

fuel pressure raising it by about 0 020 in

A low fuel level causes flat spots because of lean-out in turns and when

only enough fuel to replace that being used

The float may be made of brass stampings soldered together into an airtight

most of the common fuels, but it is always wise to check with the carburettor manufacturer if you are using a fuel other than petrol or methanol

The needle and seat fuel inlet valve controls the flow of fuel into the bowl. The

fuel flow requirement of the engine

CALCULATING FUEL FLOW REQUIREMENTS

How much fuel does your engine need? Remember we said that the best power is produced with a fuel-air ratio of 1 pound of fuel to every 12½ pounds of air (only if the fuel is petrol). Therefore if we calculate how much air the engine is pumping we can also work out our engine's maximum fuel flow requirement.

The formula we use to find the airflow in lb/hr is

$$4.38 \times \frac{D}{32.8} \times \frac{rpm}{1{,}728} \times \frac{VE}{100}$$

where D = displacement in cc, rpm = engine speed, and VE = volumetric efficiency.

At maximum torque rpm the volumetric efficiency (VE) would be 90–100% in a racing engine, and this would fall, relative to the engine's torque curve, at higher and lower rpm. If an engine produced maximum torque of 100lbf ft at 5,750rpm and 74lbf ft at 6,800rpm (maximum power rpm), the VE at 5,750rpm is, say, 95%; therefore the VE at 6,800rpm will be

$$\frac{74}{100} \times 95 = 70.3\%$$

Obviously the engine speed at which the most air is passing through the engine is at maximum horsepower rpm (Tables 4.2 and 4.3), so calculate your air flow at this engine speed.

To find the fuel flow (lb/hr), multiply the air flow by the fuel–air ratio. If the ratio is 1:12.5, multiply by 1/12.5, or 0.08. Petrol weighs 7.5lb per gallon, so to find the fuel flow in gallons per hour (gph) divide by 7.5.

TESTING FUEL FLOW CAPABILITY

How are you going to know how much fuel a needle and seat will flow? Some manufacturers have published information available, but if you are not able to uncover the particulars from the carburettor maker, you will have to measure the fuel flow yourself.

This may seem to be something of a hassle but it is a very important consideration, particularly in regard to motorcycles, as these seem to be the worst affected by fuel flow restrictions. Actually it is very easy to measure the flow to a bike's fuel bowl as you do not have to worry about fuel pump pressure.

Fill the fuel tank, and after removing the fuel bowl from one carburettor, time how long it takes to drain down to reserve. Now accurately measure how many gallons you have drained off and calculate your fuel flow in gph. Multiply this figure by the number of carburettors on the engine, and compare the answer with the fuel flow requirement for your engine. You are probably in for a surprise, many motorcycles cannot be held wide open for too long on the dyno without fuel starvation problems.

Even if you find that the needle and seat can flow sufficient fuel, you will probably find that the fuel tap will not, so repeat the test with the fuel lines removed from every carburettor, and determine the tap's fuel-flow capabilities.

Generally, it is not possible to obtain larger needle and seat assemblies for fuel for motorcycles ...

drill the fuel inlet seat's fuel discharge holes.

COLD START SYSTEM

The cold starting system provides mixture enrichment to allow starting when either the engine or the weather is cold. The system used on Weber DCOE, SU and some ...

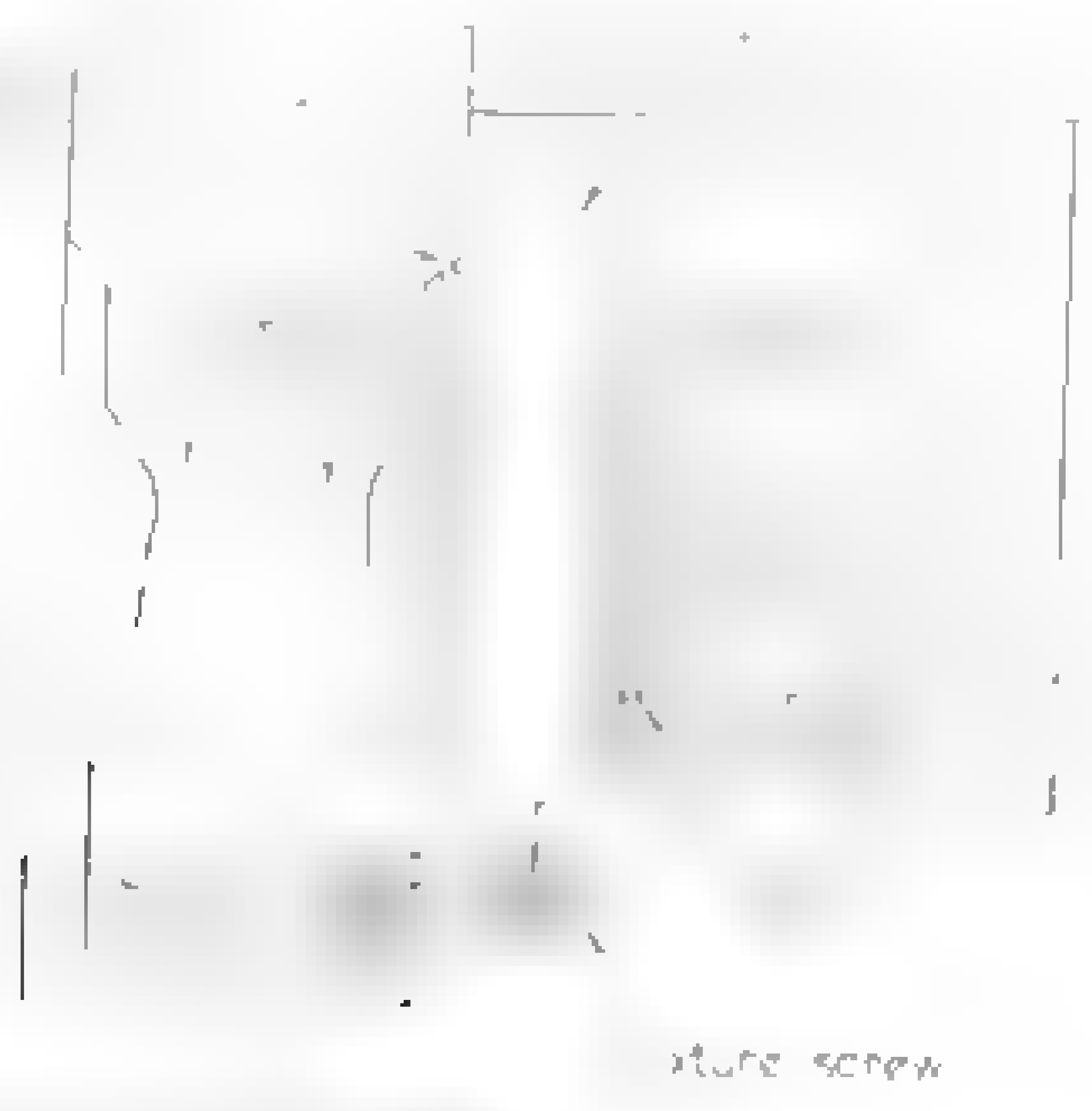
up before driving off, but this is really a good thing as engine wear and oil dilution will be reduced.

IDLE AND PROGRESSION SYSTEM

The idle system provides a rich mixture at idle and low speeds, when not enough air is being drawn through the venturi to cause the main system to operate (Figure 4.1).

When the throttle plate (butterfly) is nearly closed, the restriction to air flow causes a high vacuum on the engine side of the throttle plate. This high vacuum

Figure 4.1 Idle system operation



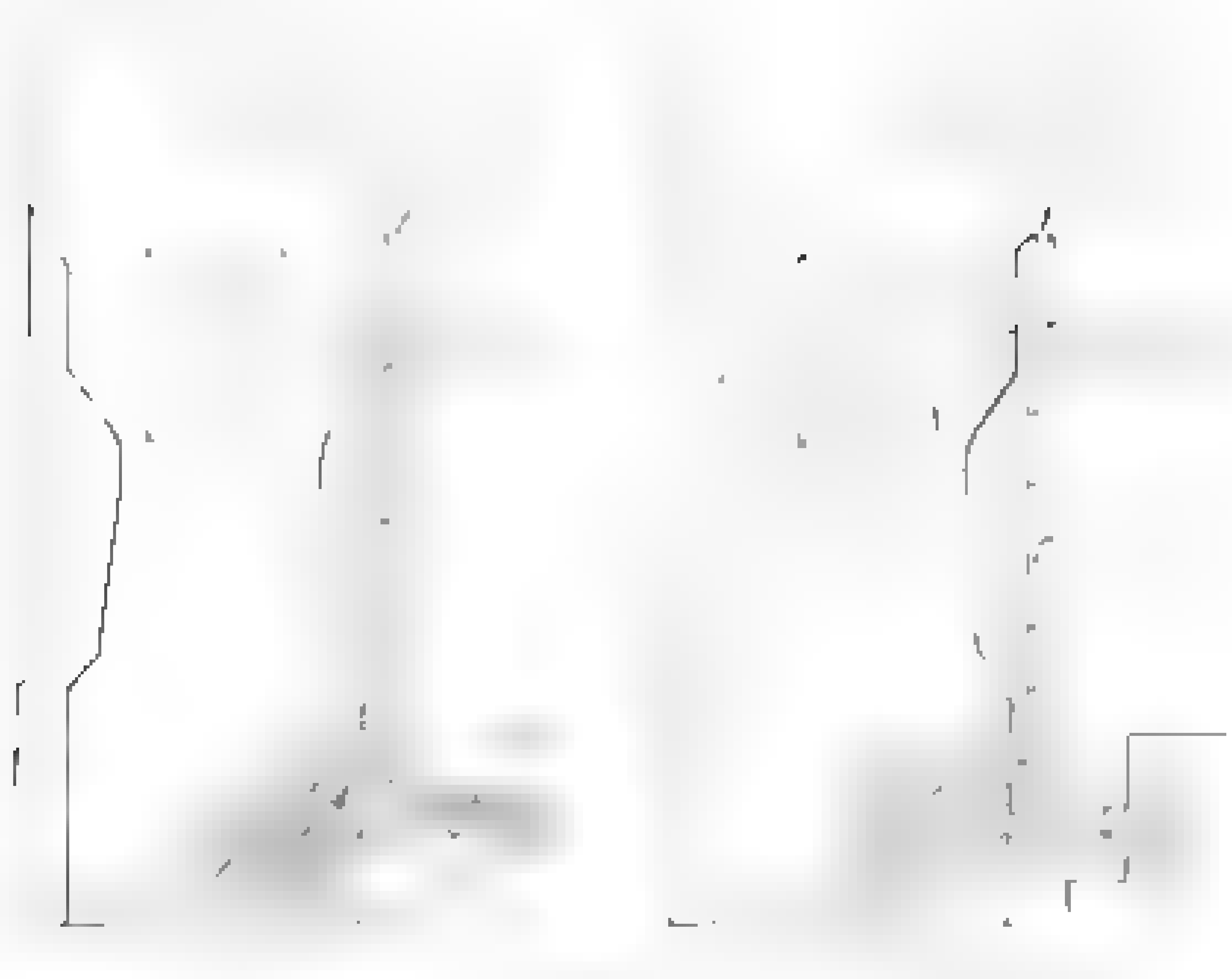


Figure 4.2 Progression system operation

provides the pressure differential for the idle system to operate. The normal air pressure (14.7psi) acts on the fuel in the float bowl, forcing it through the idle jet, and past the idle mixture screw into the manifold.

To emulsify the fuel before it reaches the mixture screw, an air bleed is included in the system. Increasing the size of the air bleed leans the idle mixture, if the size of the idle jet remains constant. Conversely, decreasing the air bleed diameter richens the idle mixture.

The progression or transfer holes are a part of the idle system (Figure 4.2) and allow a smooth transition from the idle fuel circuit to the main fuel system without 'flat spots', provided that the carburettor size has been correctly matched to the engine displacement.

As the throttle is opened wider, the progression holes are uncovered, and begin to flow fuel metered and emulsified by the idle jet. At this time fuel flow past the mixture screw decreases and gradually tapers off as the next progression hole is opened by the throttle plate.

When the throttle is opened further, the pressure differential between the idle progression holes and the air pressure acting on the fuel in the fuel bowl decreases, causing fuel flow in the idle system to taper off. Finally, the pressure is not great enough to push the fuel up to the idle jet, and the idle system ceases to supply fuel.

Fuel flow through the main metering system begins before the flow through the idle circuit is reduced, if a carburettor of the right size is being used. The main system meters fuel for cruising and high-speed operation.

As the throttle is opened and the engine speed increases, air flow through the ... increased so that the main system comes into operation, discharging fuel

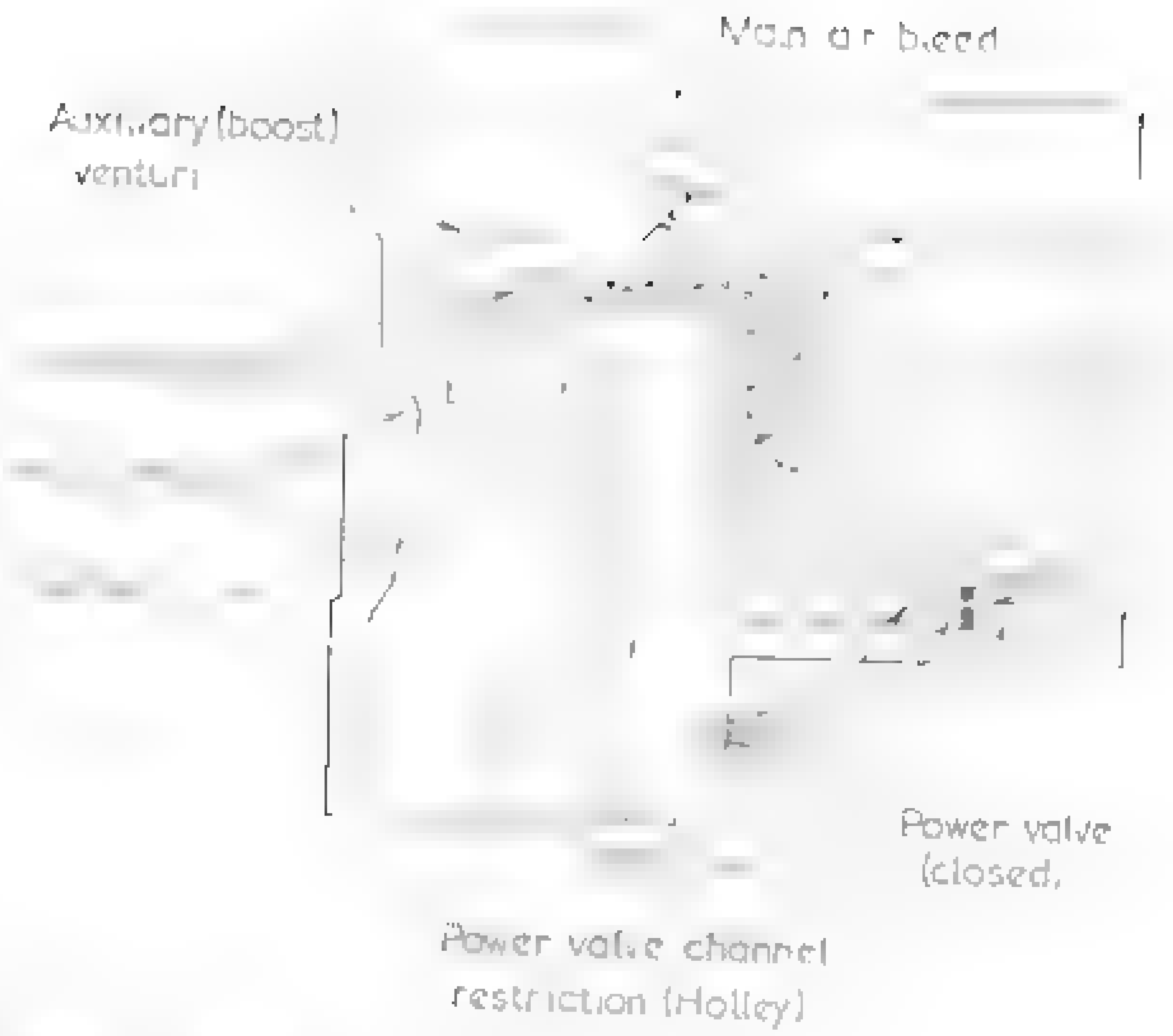


Figure 4.3 Main and power system operation

THE VENTURI

discharge nozzle (Figure 4.3)

In the internal combustion engine, a partial vacuum is created in the cylinder by

The venturi necks down the inrushing air, then allows it to widen out to the

the main jet to the discharge nozzle

Pressure drop (or vacuum) within the venturi varies with the engine speed and

nozzle This in turn gives the highest fuel flow into the engine

To compensate for various engine displacements, carburettors with a variety of venturi diameters are available to create the necessary pressure drop to bring the main fuel circuit into operation. A small venturi will provide a higher pressure difference at any given rpm and throttle opening than a large diameter venturi. This is a very important aspect of carburation, which partly explains why the biggest is seldom the best. If the signal being applied by the venturi is too weak (due to the venturi being too large), this could delay fuel discharge in the main system, causing a flat spot. If you must err when buying a carburettor, err on the small side.

The auxiliary venturi acts as a signal amplifier for the main venturi, allowing for more accurate and quicker fuel flow responses. This is important in a [illegible] engine as it allows the use of a larger, less restrictive main venturi than would normally be possible, without sacrificing throttle response.

The tail of the boost venturi discharges at the lowest pressure point in the main venturi. Thus the air flow is accelerated through the boost venturi, and because of this, the air and fuel emerging are travelling faster than the surrounding air passing through the main venturi. This has the effect of assisting fuel atomisation, and subsequently improves combustion.

MAIN METERING SYSTEM

The actual fuel metering is controlled by the main jet, the air bleed jet, and the emulsion tube.

The main jet controls fuel flow from the fuel well. An increase in diameter richens the mixture, but more is involved. The shape of the jet entry and exit, as well as the bore finish, also affect fuel flow. Carburettor manufacturers measure the flow of every jet, for high performance and low-exhaust emission applications, and number the jet according to its flow characteristics, not according to its nominal bore diameter. For this reason jets used to meter petrol should not be drilled to change their size, if you desire accurate fuel metering. An engine burning alcohol does not require such accurate metering, unless fuel consumption is a consideration, so jet drilling may be in order if large jets are not available.

The air bleed reduces the signal from the discharge nozzle, so there is a less effective pressure difference to cause fuel flow through the main jet. This allows for more fine tuning of the main metering system. A large air bleed leans the mixture, particularly at higher engine rpm.

The air bleed also introduces air into the emulsion tube, to emulsify the fuel into a lighter, frothy mixture of fuel and air. This is done to improve atomisation when the fuel is released from the discharge nozzle. Also it serves to lower the viscosity of the fuel, making it lighter, and able to respond faster to signal changes from the auxiliary venturi. This enables the main system to keep more in step with the fuel requirements of the engine.

The emulsion tube has the task of emulsifying the previously metered air issuing from the air bleed jet with the fuel coming from the main jet. Its influence is more marked at less than full throttle and during acceleration. The diameter of the emulsion tube and the size and location of holes to emulsify the fuel all affect and influence its operation. Usually, a change in emulsion tube will require a change in main jet and air bleed size.

POWER SYSTEM

Many carburettors also have a power system for mixture enrichment at heavy load

system at above normal cruising loads. The added fuel supplied by the power system is

As the load increases, the throttle must be opened wider to maintain a given speed. This lessens the restriction to air entering the engine, which in turn reduces the manifold vacuum.

The system usually employed (Figure 4.3) relies on a high manifold vacuum to hold the power valve closed. As the manifold vacuum drops (usually to around 4–6in of mercury) the power valve spring overcomes manifold vacuum and opens the valve, allowing fuel to flow from the fuel bowl through a power valve jet and into the main metering system. This increases fuel flow in the main system, effectively richening the mixture.

ACCELERATOR PUMP

The accelerator pump system complements all of the other systems and fills in for them, so as to eliminate flat spots. It does this by injecting raw fuel for a sufficient

If the throttle is opened quickly, the manifold vacuum instantly drops to a pressure approaching atmospheric, allowing the fuel to drop out of vapour and wet the manifold and port walls. (A high manifold vacuum helps to keep the fuel vaporised. Remember that an increase in altitude, resulting in a lower atmospheric pressure, lowers the boiling point of water. Likewise with fuel: lower the atmospheric pressure

to make up the deficiency by injecting more fuel into the air stream.

The accelerator pump also has to supply fuel if the throttle is opened quickly to a

FACTORS IN CHOOSING A CARBURETTOR

carburettors are suitable for performance applications. Some are just too small, others do not have metering systems with changeable jets or, if they do, the jets are not readily obtainable, and others have metering systems that allow acceptable

mixture in a high performance engine. For these reasons I usually limit myself to the use of Holley, SU and Weber carburettors for cars, and Mikuni carburettors for motorcycles. Don't get me wrong – there are other good carburettors available (eg Dellorto DHLA sidedraught), but I have found those listed to be the most easily. So

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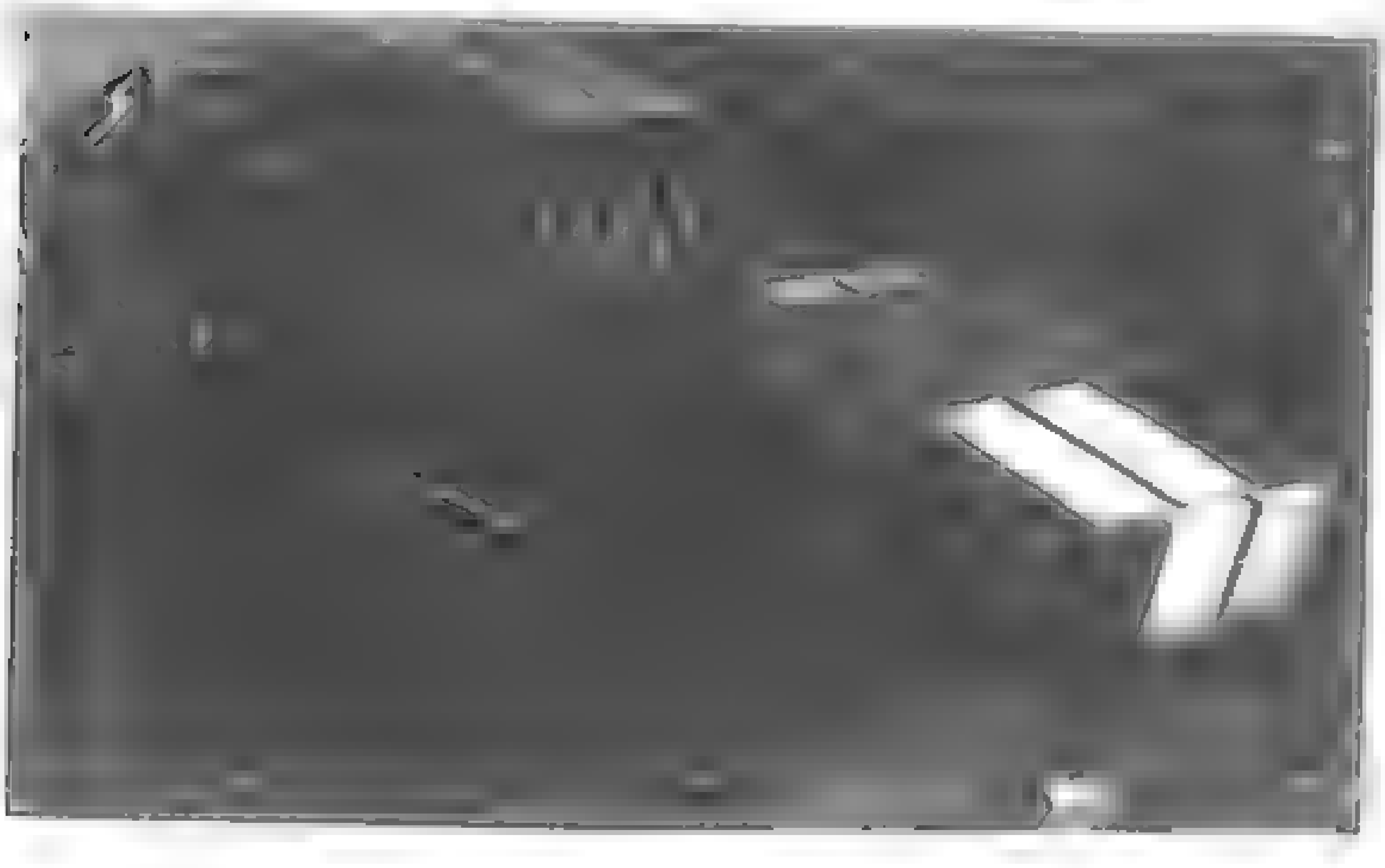


draw it up first through a straw, then a piece of [illegible] in heater hose. Did you notice the difference? How much more sucking you had to do to get the water [illegible]

[illegible]

DUAL-STAGE CARBURETTORS

[illegible]

Holley carburettors with simultaneous throttle action are for racing use only

CARBURETTOR MODIFICATION

[illegible]

the venturi bores being bored out to a larger size. Some may flow more air, but more [illegible]

air is of no value unless fuel has been mixed with it in the correct proportions, to say nothing of throttle response and metering signals. A carburettor modified along these lines will be useless in a vehicle requiring any sort of power range. It would work in a racing boat or in a stick-shift drag car with a 7,500–9,000rpm power band, but in any other application it would prove a hindrance.

The only type of modification for Holley four-barrel carburettors of which I approve involves re-calibration of the fuel metering circuits to provide more accurate metering. This modification serves to broaden the power range on a racing engine when a large carburettor is being used, by precise fuel metering right from part-open to wide-open throttle.

The problem is that racing engines produce a weaker vacuum signal due to their need for very large carburettors to provide good power at high rpm. Camshaft design also affects fuel metering, because the fuel delivery signal is weakened as the inlet valve is opened earlier and earlier, and as the valve overlap period is increased. As a result the carburettor requires more sensitivity to weaker signals, or it will excessively lean the mixture and reduce the power output at lower engine speeds.

Some feel that this problem can be solved by using a more sensitive booster venturi, but I have found the converse to be true. In a racing engine the carburettor is subject to severe reversion pulses ('spit back') and during this reverse flow period more fuel is added to the air; then as the air flows back into the engine the booster senses its flow and further fuel is added. This means that during its three passes through the carburettor, fuel has been added to the air; consequently the mixture becomes extremely rich. A booster that is more sensitive will aggravate this situation at certain engine speeds, and do little to richen the mixture at lower engine speeds where the lean condition is being experienced.

The low-speed lean condition is not, in fact, due to any insensitivity in the main metering system, but generally due to the signal being too weak to activate the idle-progression system. Changes to the accelerator pump circuit will help, but the way around the problem is to add an intermediate fuel circuit. This will fill in gaps in the Holley's fuel metering circuits and allow it to perform well at lower engine speeds, to produce a power band almost as wide as that possible when using Weber carburettors.

If you use a Holley carburettor and need a wide power range for road racing or short speedway, take your carburettor to someone who will re-calibrate it as outlined; forget about those firms who rebore the venturis.

Regardless of the type of carburettor you use, it should not be modified (except the SU) by boring the venturi or by radiusing the corners at the air entry.

If you take a careful look at the venturi you will find that it has a radiused inlet and diverging tail section. Designers work the venturi shape to create the maximum pressure drop with minimum flow losses. Frequently, when the venturi is re-bored, the basic internal shape is not changed to suit the larger internal diameter. Reboring really requires that both the entry radius and the divergent angle of the tail be changed, otherwise the increase in bore size will not increase air flow, but that will result in a less effective metering signal.

Radiusing the corners of the air entry can reduce the auxiliary venturi signal, since the inflowing air tends to turn into the main venturi sooner and take away flow from the centre of the venturi where the auxiliary venturi (booster) is located. This reduces the signal at the booster and results in main circuit fuel lean-out.

Another bad modification involves cutting down the divider wall between the primary and secondary side of a four-barrel carburettor for clearance under the hood or for improved air flow. Removing just one-quarter of an inch could upset booster sensitivity to the extent that main jets three or four sizes richer would be required, and this will not fully solve the problem

The message is to keep your fingers, your files and your grinders right away from the carburettor, particularly its mouth. Any modification in this area will almost always deaden booster sensitivity, which will generally affect fuel flow into the engine adversely

THROTTLE SHAFT MODIFICATION

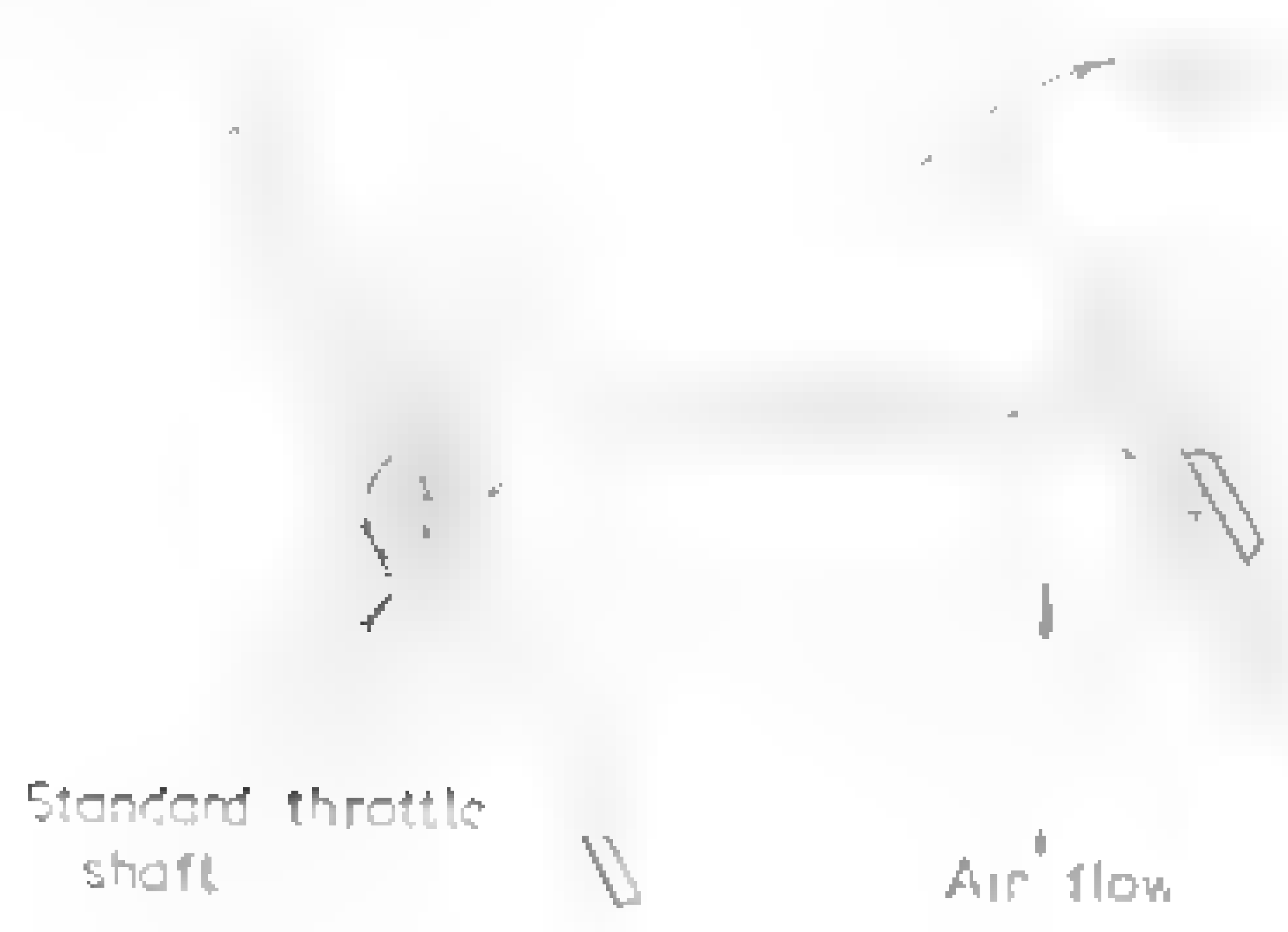
This is not to say that all modifications to improve the air flow through the carburettor are to be avoided. As I have already pointed out, you should get rid of the choke plate and shaft. That simple modification will improve the flow of a Holley by about 50cfm

The next area that you should take a look at is the throttle shaft and throttle plate screws. Most shafts are 5/16–3/8in diameter and the butterflies are attached by two screws. Some carburettors use countersunk screws, carefully trimmed so that they do not project through the shaft. However, most screws have a flat head and they usually protrude through the shaft by 1/16–3/32in. Together the shaft and screws disrupt air flow by causing turbulence and by partly blocking the carburettor bore

In racing applications, particularly if the carburettor size is limited by the rules, a useful increase in air flow (10–12%) can be gained by removing the screws and cutting [illegible] the other half slabbed so that it is no more than 1/16in thick, then ground with a radiused edge. The butterflies are attached by heliarc welding through the existing screw holes

This modification should be carried out only by a competent machine shop, taking special care not to bend or twist the very thin and weak throttle shaft If it is twisted, the butterflies will not be synchronised. The heliarc welding is also a delicate operation; again special care is necessary to ensure that the butterflies are going to open together.

Figure 4.4 Modified throttle shaft



When a carburettor modified as described is being fitted, be sure to attach the throttle return spring to the same end of the throttle shaft as the throttle linkage. If the spring is attached to the opposite end, the shaft will twist due to the opposing forces on each end.

FLOAT MODIFICATION

If you race on a speedway and use a Holley carburettor, you are probably aware of the problem of fuel surge and starvation caused by the constant side forces that push the

Wide-open throttle

[illegible]

is realised adjust the stops accordingly. The same sort of test can also

[illegible]

air stream is closer to what the engine properly requires for best hp.

fuel to the right-hand side of the float chamber, at times at a 45° angle. So steep an angle can easily uncover a main jet and allow air into the main well.

To overcome the problem remove the brass float and replace it with a modified [illegible]

will sink and flood the engine

CHOOSING THE RIGHT SIZE

Earlier it was established that the carburettor has to rely on the signal produced in its [illegible] fuel requirements are not instantly met, resulting in delayed throttle response.

Thus the displacement of the engine, the weight of the vehicle, the axle ratio and [illegible] type of carburettor that will give the best performance. To add a little extra confusion to the issue, half of the world sizes its carburettors according to the diameter of the throttle bore, while the other half rates them in accordance with how much air they flow. Therefore to know what you are actually buying, you need a little background information to understand what all the numbers mean.

All carburettors made in the UK, Europe and Japan are identified by the diameter of the carburettor bore at the throttle plate. The size may be stated in inches or millimetres. Generally, [illegible] the combination of [illegible] of tune [illegible]

$$cfm = \frac{D \times rpm \times VE}{3{,}456 \times 100}$$

where D = displacement in cu in, rpm = engine speed at maximum horsepower, and VE = volumetric efficiency at maximum horsepower.

Taking as an example the 350cu in Chevy Z-28 engine in Table 4.2, we find its air [illegible] cubic feet per minute. According to the formula, this engine would work best with a carburettor rated to flow 400–450cfm. In actual practice this particular engine will work best using a 600–650cfm [illegible] for drag racing.

The Chevy racing engine in Table 4.3 would have a peak air flow of 640.8cfm at

Table 4.2 Theoretical fuel flow in the 350 cubic inch Chevy blue-printed Z-28 (1:12.5 fuel–air ratio)

rpm	hp	torque (lbf ft)	VE	air flow (lb/hr)	fuel flow (lb/hr)
3,000	198	346	81.7	1,087.2	87
3,500	237	355	83.8	1,301	104.1
4,000	274	360	85	1,508.2	120.7
4,500	303	354	83.6	1,668.7	133.5
5,000	326	342	80.8	1,792	143.4
5,500	333	318	75.1	1,832.2	146.6
6,000	326	285	67.3	1,791.2	143.3
6,500	295	238	56.2	1,620.4	129.6

Note: the actual fuel flow will be approximately 20–25% more than the theoretical flow due to unequal distribution from cylinder to cylinder and to accelerator pump operation.

[illegible] (1:12.5 fuel–air ratio)

rpm	hp	torque (lbf ft)	VE	air flow (lb/hr)	fuel flow (lb/hr)
4,500	362	423	89.1	1,778.5	142.3
5,000	415	436	91.8	2,036	162.9
5,500	471	450	94.8	2,312.8	185
6,000	515	451	95	2,528.4	202.3
6,500	547	442	93.1	2,684.3	214.7
7,000	572	429	90.4	2,807	224.6
7,500	570	399	84	2,794.5	223.6

Note: the actual fuel flow will be approximately 20–25% more than the theoretical flow due to unequal distribution from cylinder to cylinder and to accelerator pump operation.

[illegible] would use a Holley carburettor of either 850 or 1050cfm.

On paper the air flow formula looks good, but it is of little use as the actual air [illegible] apart due to a number of factors.

In the test laboratory, carburettor manufacturers measure the air flow of each [illegible] and two-barrel carburettors, and 1.5 inches of Hg for three- and four-barrel [illegible] theoretical amount of air that two- and four-barrel carburettors will flow. A carburettor subject to a vacuum of 3 inches of Hg at wide-open throttle will naturally flow more air than one tested at 1.5 inches of Hg. A two-barrel Holley rated at 650cfm will not flow as much air as a four-barrel Holley rated at 650cfm.

To relate the two measurements we must use the form[illegible]

$$\frac{\text{cfm @ 3in Hg}}{1.414}$$

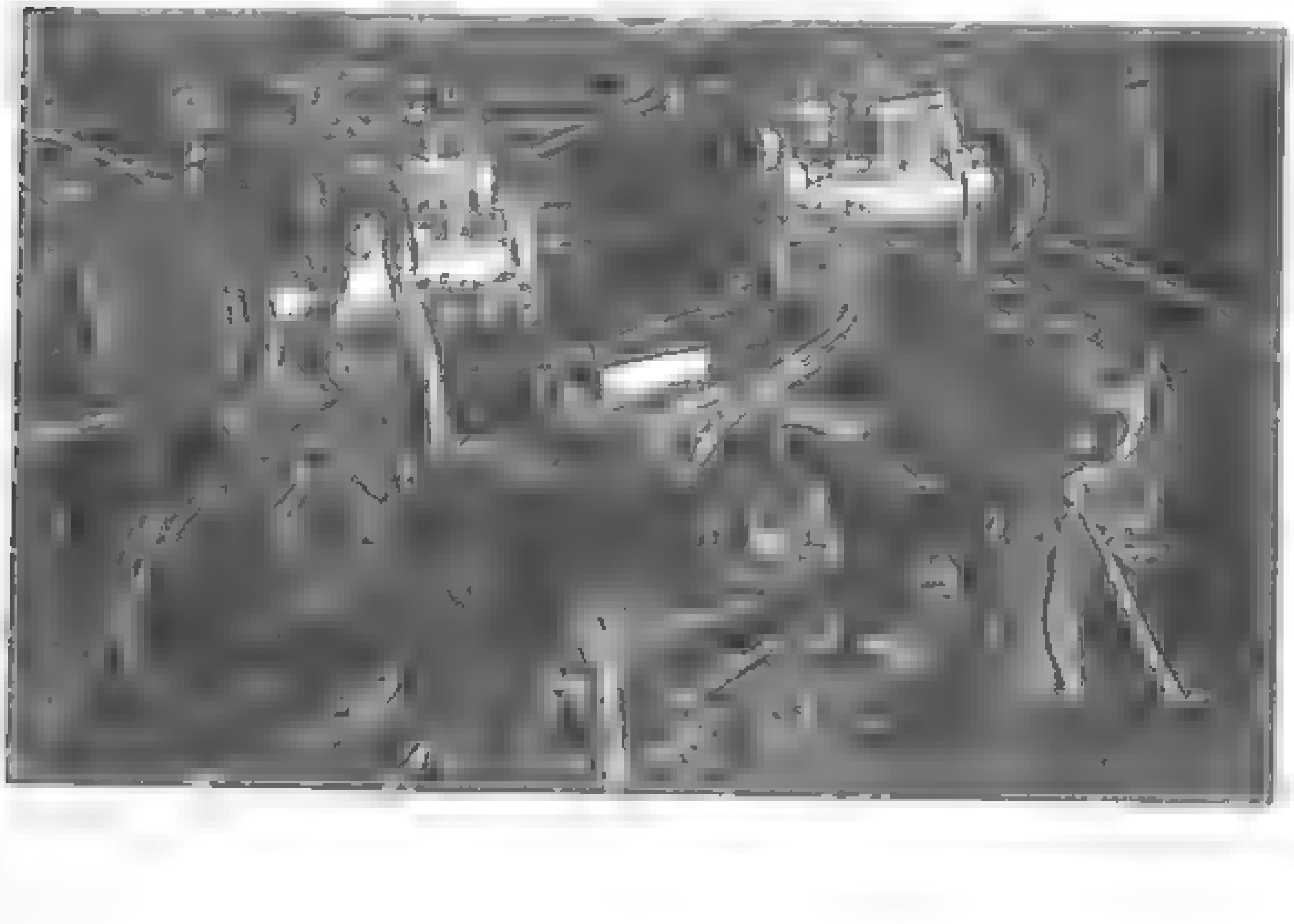
Therefore the 650cfm two-barrel will flow only 460cfm when tes[illegible] at 1.5 in Hg.

[illegible] this principle a step further, it will be found that many racing engines have a [illegible]d vacuum of only 0.5 inches of Hg at wide-open throttle, not the 1.5 inches at [illegible] l carburettors are flow teste[illegible] sion of 0.5 inches of Hg the [illegible] will equal the cfm at 1.5 inches of Hg divided by 1.735. Therefore a 1,050cfm carburettor will flow approximately 605cfm when fitted to a racing engine with a [illegible] vacuum of 0.5 inches of Hg.

[illegible] the test bench the carburettors are subjected to [illegible] constant vacuum for air flow measurement purposes, but in practice a carburettor fitted to an engine has to contend with reverse flow (or reversion) pulses, which also cause the [illegible] r flow capacity of the carburettor to be reduced. Racing engines produce a stronger reversion pulse than stock units, due to the use of wild camshafts that cause more reverse flow in the inlet tract because of early inlet valve opening and lots of valve overlap. Manifold de[illegible] also influences the intensity of the reverse pulse.

Another aspect of carburettor air flow not considered in the flow room is the on and-off-type of flow actually experienced by the carburettor when the inlet valve opens then closes. If the inlet valve was held wide open for 1 minute, carburettor air flow

Fitted to an isolated runner manifold these 48 DCOE Webers are adequate up to abo[illegible] 240hp. However, used on this two-plane V8 manifold, they can flow enough air to more than double that figure



would be constant for that minute, but in use the inlet valve may open and close 4,000 times during that minute, serving to reduce air flow. Obviously a tap turned on for a minute will flow more water than one that is turned on and off 4,000 times in a minute.

The more cylinders to which a carburettor venturi is flowing air, the more [illegible]

[illegible]

Escort rally engine, you will find that it breathes through two 48DCOE Webers, each [illegible] 240hp from 2,000cc, and a power band compatible with international rally events. [illegible] carburettors will flow enough air to produce in excess of 525hp.

ISOLATED RUNNER MANIFOLD

The RS 1800 uses what we call an 'isolated runner' manifold (Figure 4.5), with one carburettor throat directly connected to each cylinder. This means that each cylinder is totally independent of and isolated from all of the other cylinders. Fuel distribution problems and charge robbing is eliminated, improving fuel economy and power output. An induction arrangement of this type is commonly found on road racing engines, European performance cars and all motorcycles.

From the aspect of horsepower output there is not a lot of difference in the peak power output of V8 engines using four Webers on an isolated runner manifold, or two four-barrel Holleys on a plenum ram manifold. If fuel distribution is given proper attention, the latter would produce more power (at times up to 7% more). For this reason Holleys are usually preferred for drag racing, but for road racing, where a wide power band and good throttle response are more important, an isolated runner manifold and Webers are a combination hard to beat.

An important consideration when you go out to buy an isolated runner manifold is [illegible]

Weber chokes (venturis) too small?

[illegible]

Figure 4.5 Isolated runner manifold

[illegible]

TWO-PLANE MANIFOLD

Most V8 engines in production use a two-plane manifold (Figure 4.6). Originally this [illegible] other half and the second plane, assuming a 1–8–4–3–6–5–7–2 firing order. This [illegible] the manifold is of a good design.

Because there is less air mass to activate on each inlet pulse, throttle response is usually quicker and mid-range power is improved, as compared to a single plane manifold. The division of the manifold into two planes is, however, a mixed blessing as it causes a flow restriction at high rpm because only half of the carburettor flow capacity is available on any intake stroke.

For this reason the divider can be reduced in height or sometimes removed completely, to make more carburettor flow capacity available, top-end performance is [illegible] can be a problem, but not to the extent of single plane designs.

The two plane manifolds fitted in normal production are generally of very poor design, with many tortuous corners and restrictive passages that retard air flow. For

most V8 road vehicles I recommend the use of either an Edelbrock or Weiand two- [illegible] shift that could result in port misalignment.

I like this type of manifold for the good throttle response it is able to produce [illegible]

SINGLE-PLANE MANIFOLD

[illegible] barrel carburettors, or a single two- or four-barrel carburettor.

Figure 4.6 Two-plane V8 manifold

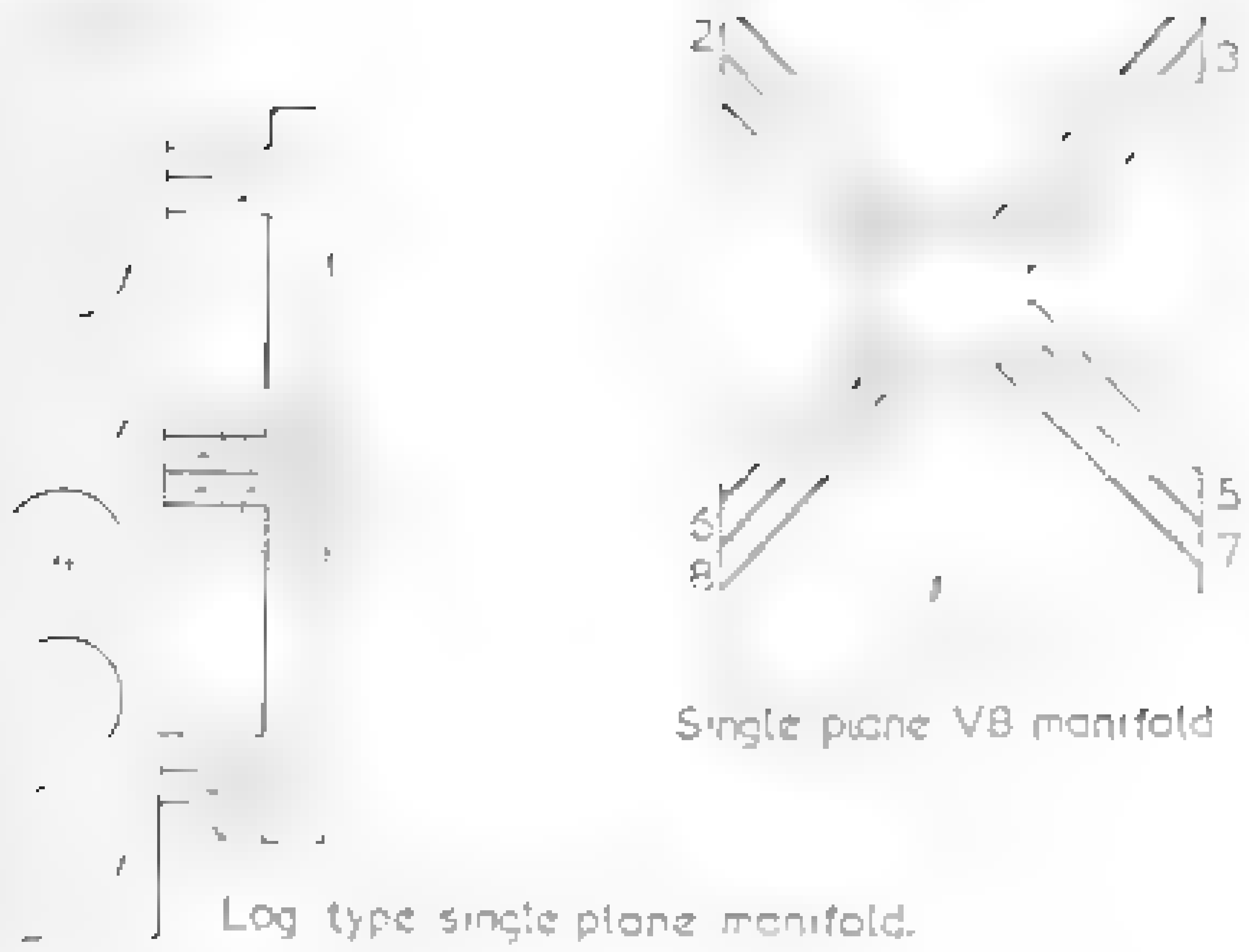


Figure 4.7 Single plane manifolds.

The main problem with the single-plane design is the tendency for one cylinder to rob mixture from another. However, by careful design, this deficiency can, and is, being overcome by manifold manufacturers.

In general I prefer to keep well away from single-plane manifolds for in-line engines as they are ideal for the isolated runner set-up. For sports engines the single-plane manifold will work reasonably well, providing that the manifold is of a 'log' design (Figure 4.7). To the eye this type may not look as racy as a manifold with nice curves, but it will assist in mixture distribution and will also help prevent the fuel dropping out of suspension and wetting the manifold and port walls.

The pockets produced at the ends of the log manifold and beneath the carburettor tend to form a soft air cushion that helps keep the mixture atomised as it makes a change in direction. Also, because the shape of the end ports more closely resemble the shape of the inner ports, mixture distribution is equalised. A good rule to keep in mind if you are using a single-plane manifold and you desire reasonably equal mixture distribution is this: where possible, divide the mixture before a change in direction occurs.

There are a number of good, single-plane V8 manifolds available, but in general I stick to the Holley Street Dominator for all engines in sports tune, and engines up to 320cu in in semi-race tune. The Edelbrock Victor Junior is suitable for motors larger than 320cu in, if in semi race tune. For race engines, the Holley Strip Dominator and Edelbrock Victor are a good choice.

Because throttle response and low-end power is not so good, it is best to use a vacuum secondary Holley of around 600cfm for street engines up to 350cu in and a vacuum secondary of 780cfm for larger engines. Race engines work well with a 750cfm mechanical secondary double-pumper up to 315cu in, an 800cfm up to 350cu in, and a 1,050cfm above 400cu in.

Offenhauser have tackled the problem by offering a dual-port, single plane manifold. A small-passage and large-passage manifold are stacked one on top of the other in a single casting. The small ports are connected to the primary side of the [illegible] and gives good throttle response. The larger ports connect to the secondary barrels of the carburettor for high rpm operation.

There are several features that the dual-port manifold can boast. The small [illegible] the secondary passages. This serves to increase the secondary air density, thus improving high-speed performance.

The small passages also lift mid-range performance in this way. When the [illegible]

PLENUM RAM MANIFOLD

All-out drag cars and racing boats commonly use a plenum ram manifold (Figure 4.8) with either one four-barrel, or more commonly dual four-barrel, carburettors. This manifold is very similar to the isolated runner manifold, with a plenum chamber added [illegible] helps to dissipate the strong pulsing, common to the isolated runner design, so less pulse enters the [illegible] will not work below 7,000rpm.

For Chevrolet engines I prefer the Holley Pro Dominator manifold. This type has [illegible] to carry out modifications it is frequently necessary to saw the chamber in half. Then, after any grinding and filing, it has to be welded together – all very expensive and time-consuming.

CHOOSING A MANIFOLD

When you go out to buy a manifold there are several things to look for and to keep in mind. First, you want to buy a manifold that will, if possible, fit the carburettor(s) that you wish to use, without an adaptor. Generally, adaptor plates restrict the air flow by creating undue turbulence, because they attempt to change the air flow path too quickly, the plate being only ½–1in thick.

Many of the adaptors made to mount the large Holley 4500 carburettors (1,050 and 1,150cfm) on standard four barrel-pattern manifolds restrict the air flow

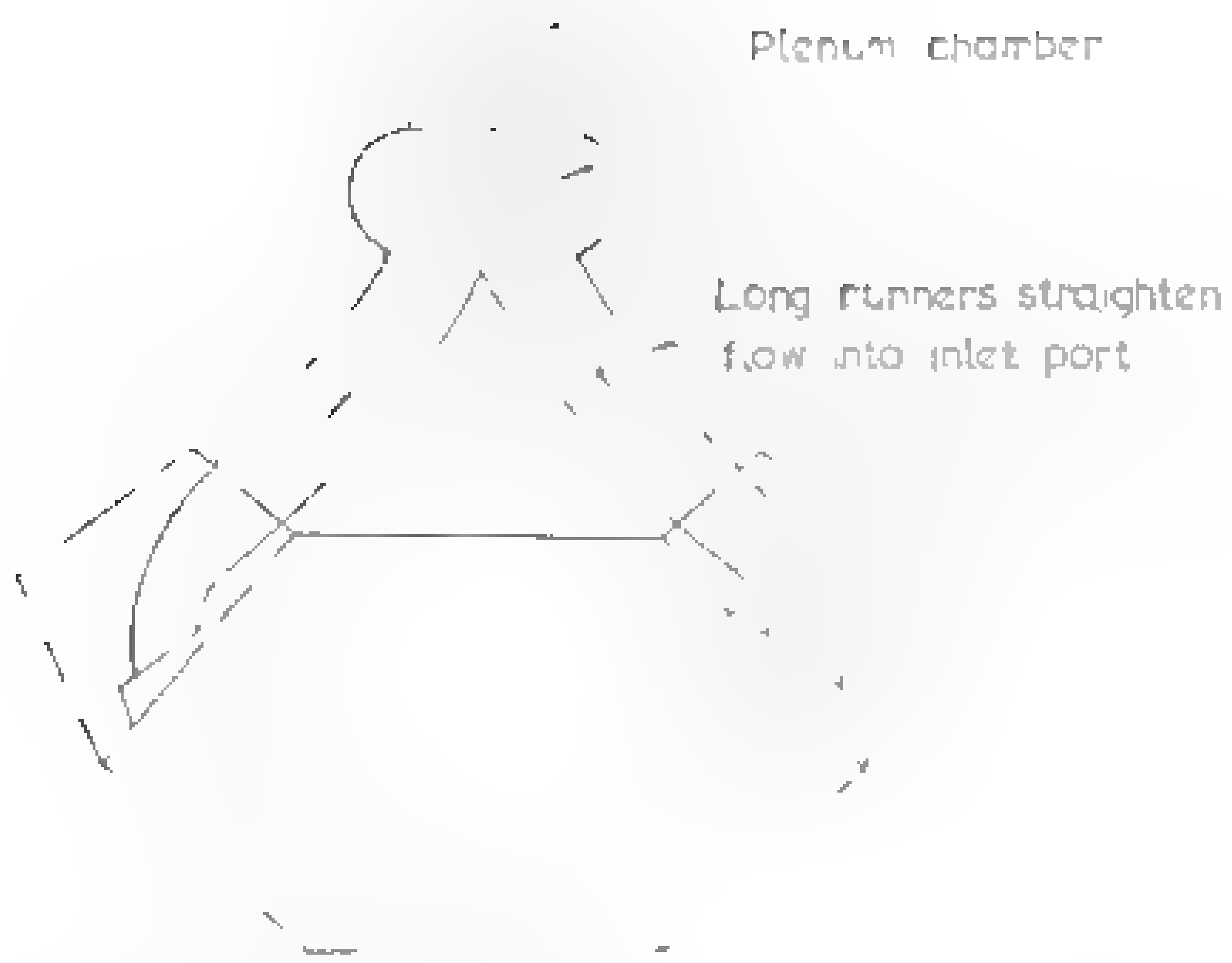


Figure 4.8 V6 plenum ram manifold.

of the carburettor by 100–250cfm, which effectively takes away any advantage of the larger carburettor.

Take an inlet manifold gasket with you to check how the manifold ports match up. As most stock gaskets have port holes much larger than necessary, to allow for port misalignment, it will be necessary to make up a gasket with the exact port and stud locations of your cylinder head. If your engine is a V6, V8, flat-four, etc, you will need two gaskets; be sure to mark each left or right.

Try to find a manifold that matches as closely as possible to the gasket as this will save some work when the manifold and head are matched. Additionally, look at the ports to check their smoothness; high spots ('dingle berries') can be ground out, but depressions can be a problem if a suitable epoxy filler is not available.

You should also note the 'join' in the manifold, which is where the core boxes were joined when the manifold was cast. At times the core and core boxes are not correctly aligned, resulting in half of the port being offset from the other half. Choose a manifold with little or no evidence of core shift.

MANIFOLD MODIFICATION

After you have bought your manifold, try fitting it to your engine before you modify it in any way, as you may have to return it if it does not fit. Some V6 and V8 manifolds will not fit with non-standard distributors, on the other hand a couple will not fit with the stock distributor. Due to the large variety of exhaust headers available, a few manifolds will not clear the headers on in-line engines.

When you are sure that the manifold will clear everything, match up the manifold runners with the inlet ports and match the carburettor bore to the manifold

entrance. The only exception to this is with Edelbrock Streetmaster manifolds, as these

MINIMISING FUEL DISTRIBUTION PROBLEMS

variation in the quantity of fuel reaching each cylinder.

Generally, V8 engines using single-plane and low profile cross-ram manifolds have

performance manifolds are quite good. So too are the plenum ram manifolds.

variations from cylinder to cylinder.

Before we delve into manifold modifications to assist in achieving equal fuel distribution, there are a couple of principles that should be considered.

on speedway machines where the cornering force may be reasonably constant

In production engines, exhaust heat or hot engine coolant is circulated through the manifold floor directly below the carburettor. The heat will cause some fuel to vaporise, but we may want to remove the heat in semi-race and full-race engines to improve high rpm air density. In this instance we may have to sacrifice distribution uniformity at lower engine speeds to ensure peak power at maximum rpm.

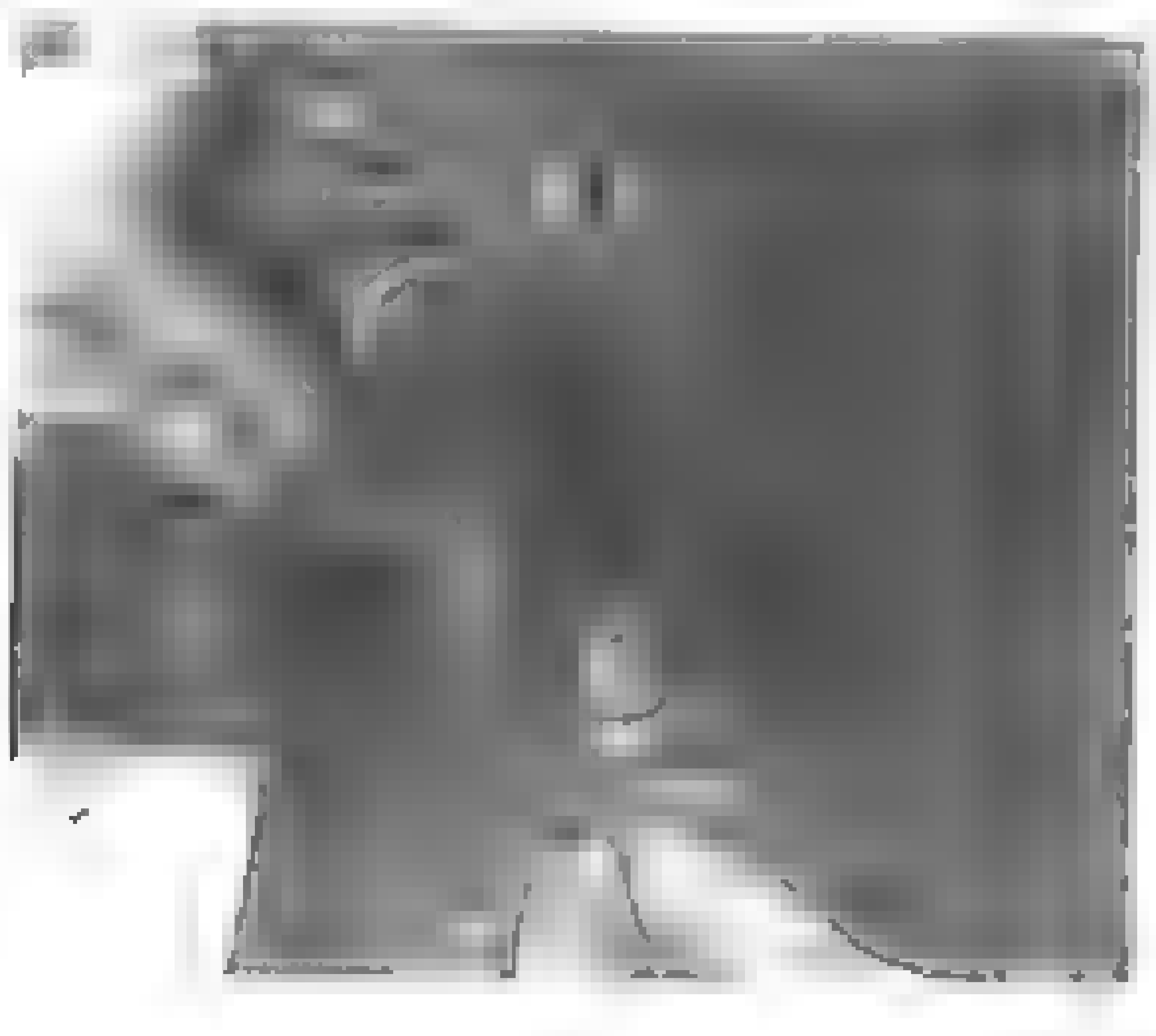
If hood clearance is not a problem, raising the carburettor using a 1–1½in spacer may assist distribution in two ways. First, because the carburettor is further removed from the floor of the manifold, the fuel/air charge will have less directional effect imparted to it as it leaves the carburettor. This will allow the charge to flow from the carburettor, around the corner of the manifold opening and into the ports. If the carburettor is mounted very close to the manifold (as is often the case) the fuel air [illegible] into the manifold and throws the fuel out of suspension and on to the floor of the manifold. The fuel is much heavier than the air so it cannot change direction quickly, therefore it continues straight on until it hits the manifold floor.

At small throttle openings the spacer also promotes better distribution by reducing the directional effect imparted to the fuel/air mixture by the angle of the throttle plate. This can be a serious problem in the upper plane of two-plane manifolds, causing two cylinders to run very lean. After the spacer (riser) is fitted, distribution will be improved markedly.

After ensuring that the manifold is horizontal, a check must be made on the actual fuel distribution from cylinder to cylinder. The only really accurate way to determine this is by fixing a lambda sensor into each header tube to show which cylinders are rich and which are lean.

As a back-up, the exhaust gas temperature measurement system is also used. It is less sensitive than chemical analysis, as it does not take into account exhaust temperature variations from cylinder to cylinder, due to differences in exhaust scavenging. However, it is a particularly valuable test in the dyno room, to determine the heat level of a given cylinder, so as to avoid engine damage due to an excessively lean mixture condition.

Careful inspection of the spark plug tips will give some indication of mixture distribution problems in engines allowed to run leaded fuel



Obviously at the race track, or on the road, you are not able to use either system. In this case your eyes become the test instrument as you rely on the appearance and colour of the spark plug electrodes and porcelain to tell the story of fuel distribution equality.

At the very best this system is only a rough guide, but it will provide us with valuable information if approached correctly. The engine must be in good condition, broken or worn rings and worn valve guides will distort the readings very badly. Also the spark plugs must be of the correct heat range for the engine.

Actually, the colour of the porcelain insulator nose is not as important as some would suppose, as it may not be possible to colour the plugs using certain types of fuel unless the mixture is extremely rich. Added to this it can take many, many miles for the insulator nose to colour, so you will appreciate that there is a good deal more to plug-reading than merely examining the colour of the insulator.

It takes practice and a proper magnifier of 4x to 6x power to pinpoint fuel distribution equality. The things to look for that indicate certain operating conditions are indicated in Table 4.4. You will note that all of the plug end actually exposed to the combustion flame is examined and read, not just the insulator nose.

Table 4.4 Checking fuel distribution by spark plug reading

Spark plug mixture condition	Indications
Normal – correct mixture	Insulator nose white or very light tan to rust brown Little or no cement boil where centre electrode protrudes through insulator nose Electrodes not discoloured or eroded
Fuel fouled – rich mixture	Insulator nose dark grey or black Steel plug shell end covered with dry, black soot deposit that will easily rub off
Overheated – lean mixture	Insulator nose chalky white or may have satin sheen Excessive cement boil where centre electrode protrudes through insulator nose. May be milk white or meringue-like Centre electrode may 'blue' and be rounded off at edges Earth electrode may be badly eroded or have molten appearance.
Detonation – lean mixture	Insulator nose covered in tiny pepper specks or maybe tiny beads of aluminium leaving piston crown Excessive cement boil where centre electrode protrudes through insulator nose Specks on steel plug shell end

For the plug reading to be accurate it will be necessary to run the engine at full throttle and maximum speed on the track (or road), then immediately cut the engine dead. If you allow the engine to slow down as you bring the vehicle to a stop, the plug reading will be meaningless.

Table 4.5 Checking fuel distribution by combustion chamber, exhaust valve and piston crown colour

Mixture condition	Indication
Normal	Dry, dark, hard carbon in combustion chamber and piston crown Light tan to rust brown exhaust valve
Rich	Dark soft carbon in combustion chamber and on piston crown. Carbon may be wet or have wet stains Dark exhaust valve
Lean	Grey to white deposit in combustion chamber and on piston crown Oil may be burned a very dark brown under piston crown. Called 'death ash' if burned to black ash White chalky colour exhaust valve

A tuner must use everything at his disposal to check the distribution from cylinder to cylinder. Usually, race engines are frequently stripped for inspection and periodic replacement of parts. This is an ideal time to read the combustion chamber and also the exhaust valves and the top of the piston. Naturally if the engine is burning a lot of oil through being overdue for a rebuild, a reliable reading will not be possible (Table 4.5).

SINGLE-PLANE V8 MANIFOLD 'FIXES'

Having determined which cylinders are rich, normal or lean, we must then set about correcting the problem. Usually, single-plane V8 manifolds cause us the most problems. As this design is very popular in high performance and racing applications, we will concentrate on 'fixes' to overcome distribution deficiencies in manifolds of this type.

The distribution situation can at times be improved by staggered jetting of the carburettor. If you find two adjacent cylinders (ie 1 and 3, 5 and 7, 2 and 4, or 6 and 8 – or for Ford engines 1 and 2, 3 and 4, 5 and 6, or 7 and 8) that are both either rich or lean, the condition may be corrected by changing the jets in the barrel of the carburettor feeding those cylinders. This will sometimes correct the distribution, but if you find little improvement after several jet changes, you can assume that manifold modifications are necessary.

Before you alter the manifold it is advantageous to determine if the manufacturer has incorporated changes in later versions of that same basic manifold to fix a distribution problem. You will find that manifold manufacturers are continually updating their products, so by studying the refinements made by the manufacturer and incorporating these in your manifold, you can save yourself a lot of time experimenting to find a suitable 'fix'.

Earlier I mentioned that there is always some liquid fuel running around the floor of the manifold. If some of this fuel can be directed into a lean cylinder, or prevented from entering a rich cylinder, the power output of the engine will be improved and the fuel consumption will decrease.

Most racers are after more power, but few are concerned about fuel consumption. However, in reality many long-distance events on road circuits and super speedways have been won and lost on this point. Tuners who work really hard to equalise fuel distribution have been able to get 10–15 more laps out of each tank of fuel on the super speedway than their competitors, on the same number of gallons of fuel.

There are two very simple modifications that will correct the majority of distribution deficiencies: fuel dams and fuel slots (Figure 4.9).

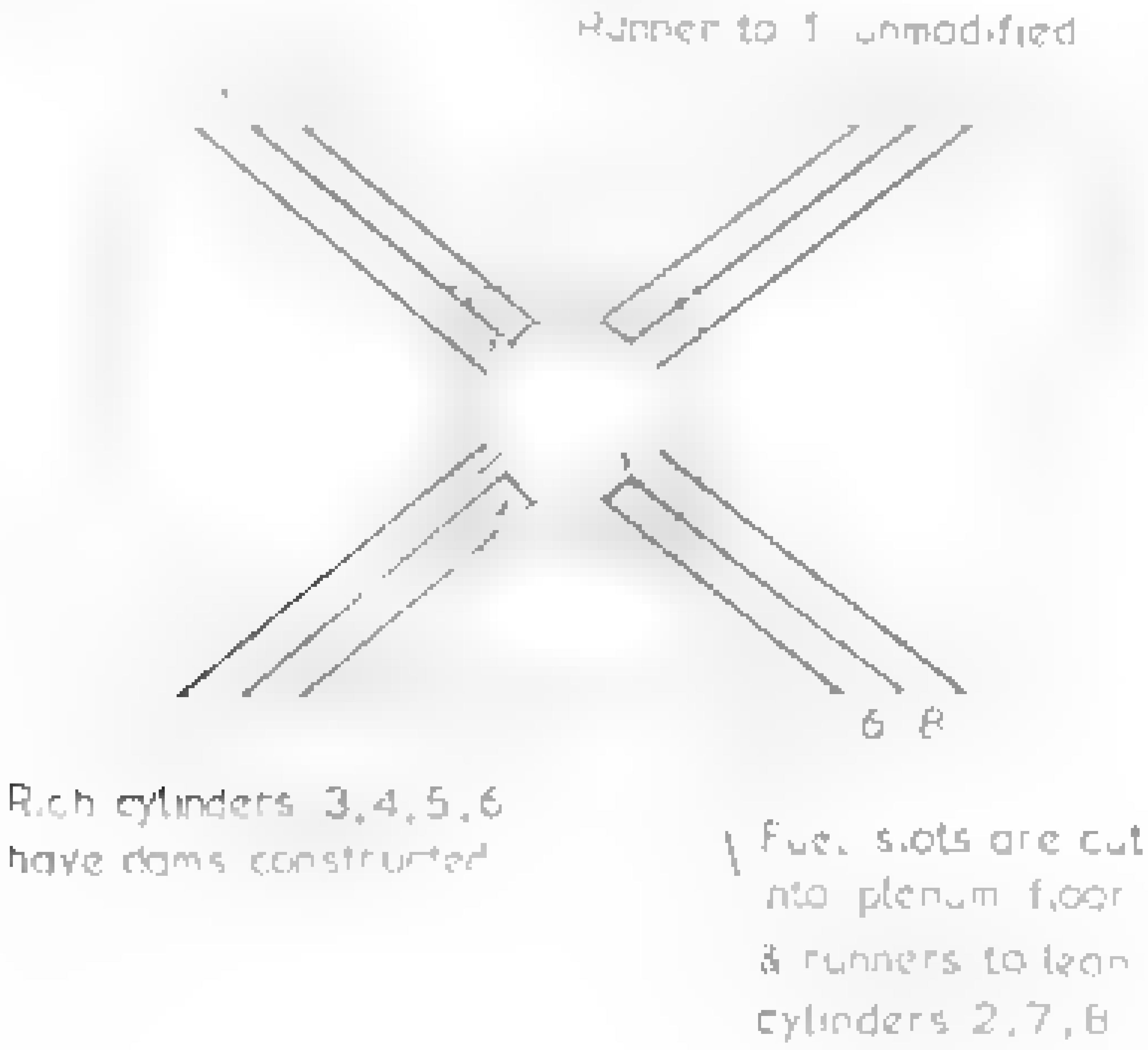
A dam can be constructed of 1/8in rod epoxied to the plenum floor (use a fuel-resistant epoxy). It may be placed across the entry of just one runner or a pair of runners, to reduce the amount of liquid fuel reaching those cylinders. After this modification it is sometimes necessary to fit leaner carburettor jets, as the cylinders without the fuel dams may become too rich.

Fuel slots can be utilised in conjunction with dams or they may be beneficially used just by themselves. Such slots are used to channel liquid fuel to the individual manifold runners that need it. The slots are cut in the floor of the plenum chamber and runner with a high-speed 1/4in spherical cutter to a depth of 1/8–3/16in. To ensure that the fuel reaches the desired cylinder, the slot must extend approximately 1in past the entry of the runner into the plenum chamber, and 1 1/4–1 1/2in down the throat of the runner itself.

You are probably wondering how the directing of liquid fuel to a lean cylinder, or preventing it from entering a rich cylinder, will improve power output. It is true that liquid fuel contributes little in the way of actual power because of its very slow burning characteristics, but more is involved.

Normally, to prevent damage in a lean cylinder it is necessary to increase the carburettor jet sizes so that this cylinder receives sufficient fuel. As a result the power

Figure 4.9 Manifold modified to improve fuel distribution



output of the engine will be reduced as the rest of the engine will no longer be receiving a fuel/air mixture for best power; the other cylinders will now be too rich.

However, by channelling liquid fuel to a lean cylinder, the fuel/air mixture in the other cylinders will be virtually unchanged. The liquid fuel passing into the lean cylinder will not do much to lift its power output to equal that of the other individual [illegible] piston or valve burning and serious detonation. With the lean cylinder taken care of, it is [illegible] are receiving the best fuel/air mix for maximum power. This results in a power [illegible] and be less prone to plug fouling.

Moreover, restricting the flow of liquid fuel to a rich cylinder raises the engine's [illegible] power stroke. The result is a loss of power.

ATMOSPHERIC CONDITIONS AFFECT JETTING

You already know that air is less dense at 5,000 feet than it is at sea level, but did you realise that prevailing atmospheric conditions could reduce the air density at sea level to approximately the same density as air at an altitude of 5,000 feet?

Since the temperature, humidity and barometric pressure all affect air density, it [illegible] from hour to hour, and this will influence the power output of the engine. Under normal circumstances the change in air density from hour to hour is of little or no consequence to the average enthusiast, but the racing engine tuner seeking as much power as possible, or wanting to prevent burned pistons and valves, has to take the present air density into consideration before each and every race or during a rally, where large changes in altitude and climatic conditions are experienced.

When the air density decreases, it reduces the amount of oxygen inducted into the cylinders, and the mixture becomes richer. Conversely, an increase in the air density increases the quantity of oxygen entering the motor, so there is a corresponding leaning of the fuel/air mixture. To compensate, it will be necessary to fit richer or leaner carburettor main (and at times idle) jets.

Remember when compensating for a change in air density that the change in density also affects the pressure exerted on the fuel in the float bowl. Therefore a decrease in relative air density (RAD) will automatically lean the fuel/air mixture to a degree, because of the lower air pressure.

This means that you do not fit 5% smaller jets when the RAD drops by 5%. Usually I reckon that a 12% change in RAD requires a 5% change in fuel jet size.

To make sure that the correct change in jet size is made, you first have to know the relative air density at the time when the carburettor was originally jetted to deliver peak power. If the engine was set up on a dyno, the air density will usually be stated on the dyno sheet, so you can work your calculations from that figure.

If the engine was not tuned on the dyno, or the dyno jetting is found to be all wrong (this can happen), you have to set about finding the ideal jet sizes from first [illegible]

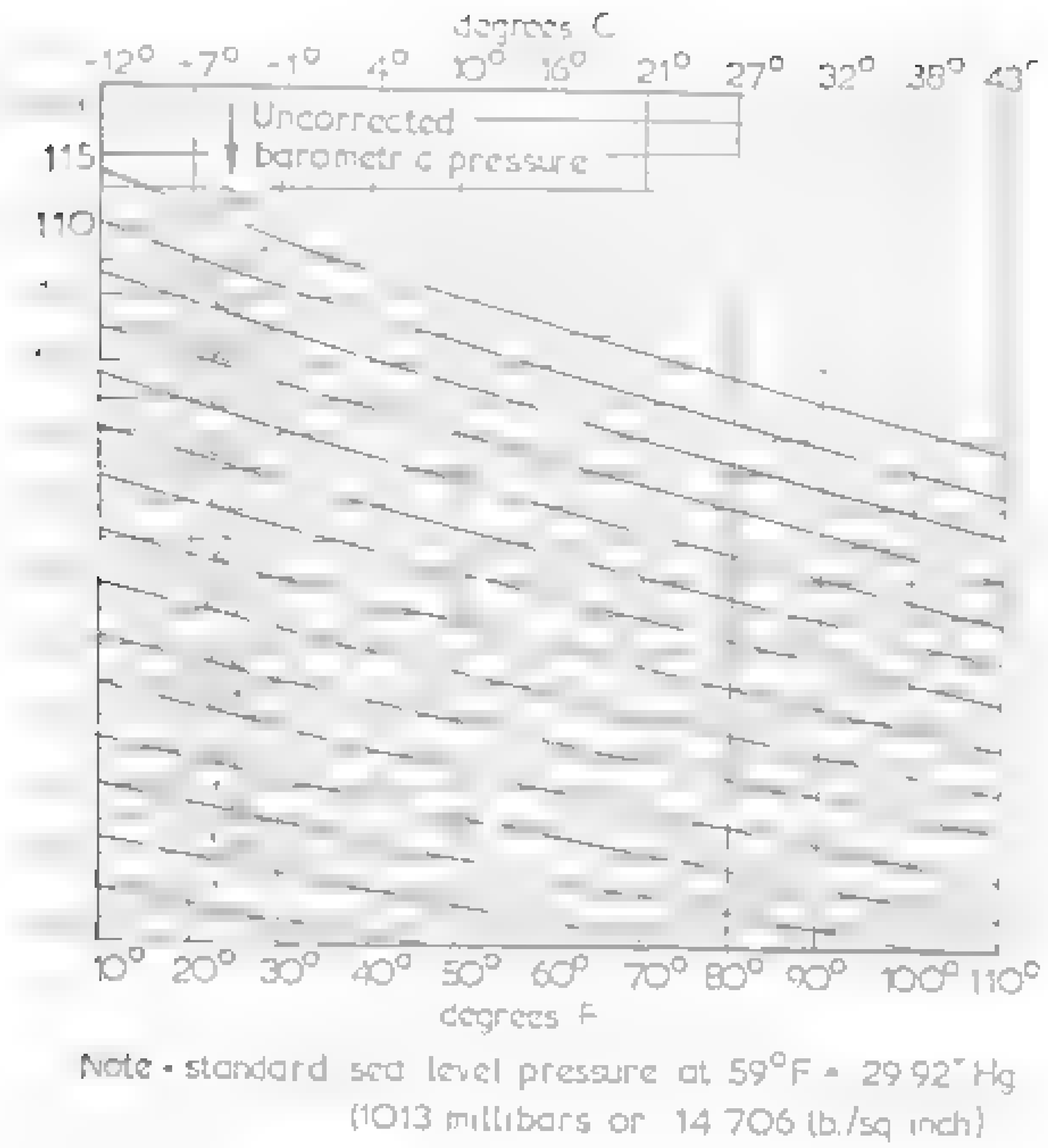


Figure 4.10 Relative air density chart.

principles. When you find what jets give the best performance, make a note in your tuning diary of the jet size and the relative air density, for future reference.

The relative air density can be worked out (Figure 4.10) providing that you know the air temperature and the uncorrected barometric pressure. Relative air density meters are available and these give a direct percentage density reading.

There is another factor involved and unfortunately this cannot be read off the relative air density graph or meter, but as its influence affects the true air density, we have to take it into account to be completely accurate.

The effect of the humidity on the density of the air is small except when both the temperature and the relative humidity are high. Water vapour has weight and as such combines with the weight of the air to distort the true weight or density of the air. Think of it in this way: you are the air and your clothing is water vapour. With clothes on you are going to exert more pressure (weight) on the bathroom scales than your true undressed weight. To find your true weight you have to subtract the weight (or pressure) exerted by your clothes. Similarly, when we want to find the true air density, we have to subtract the pressure exerted by the water vapour.

If you look at Table 4.6 you can see that the pressure exerted by water vapour at [illegible] is 1.93 inches of Hg. If [illegible] the true air pressure is only 28.07 inches of Hg, or a drop of 6.4%. Therefore the fuel/air ratio will be 6.4% rich if the jets are changed to suit the uncorrected air density.

Table 4.6 Humidity saturation pressure and percentage

Temperature		Saturation pressure		Saturation [illegible] of water
(°F)	*(°C)*	*in Hg*	*millibars*	
0	−17.8	0.038	1	0.12
20	−6.7	0.103	3	0.33
40	4.4	0.247	8	0.83
60	15.6	0.521	18	1.7
70	21.1	0.739	25	2.5
80	26.7	1.03	35	3.3
90	32.2	1.42	48	4.7
100	37.8	1.93	65	6.5

Usually, the amount of water vapour is less than the amount indicated in the column headed 'Saturation pressure', as this assumes a relative humidity of 100%. (Relative humidity compares the amount of water vapour present with what the atmosphere is capable of holding.)

To find the true air pressure use the formula

$$\text{CAP} = \text{UMP} - \left(\frac{\text{SP} \times \text{RH}}{100}\right) \text{ inches of Hg}$$

where CAP = corrected air pressure; UMP = uncorrected barometric pressure, ie read straight off the barometer; SP = saturation pressure (from Table 4.6), and RH = relative humidity

Once the corrected air pressure had been calculated, the true relative air density can be read straight off the relative air density graph (Figure 4.10)

If you are using a relative air density meter, the percentage reading must be corrected using the formula

$$\text{corrected RAD} = \text{RAD reading} - \text{S\%} \times \frac{\text{RH}}{100}$$

where S% = saturation percentage of water (from Table 4.6) and RH = relative humidity.

The one thing that you must be sure to do if you wish to be successful in tuning your motor according to the relative air density is to keep complete and accurate notes. If you find that the engine works best with 150 main jets and 10° initial spark [illegible]

when the air density is 90%, be sure that you make a note of the fact in your tuning diary. Then on each occasion that the air density is again 90%, you will know exactly what size jets and how much spark advance to use to obtain peak performance.

At another location the relative air density might be 98%, so armed with the information in your diary you know that you should try the next larger size of main jet. Maybe it will be the correct size, maybe it will not. There are no hard and fast rules here; no two engines respond to air density changes exactly alike. Usually, small displacement, high rpm engines, in a high state of tune, are the most affected by a change in air density.

MANIFOLD AIR LEAKS

The mixture ratio is also affected if there are any air leaks, so you must be very careful to seal the manifold to the head, and the carburettor to the manifold, using the correct gasket and the right type of gasket cement. This may seem rather trivial but you would be amazed at the number of tuners who use Silastic to seal the manifold gasket. Silastic is excellent as an oil sealant but it is not petrol resistant, therefore it should not be used anywhere in the induction tract. I recommend the use of Permatex No 3 in this area.

Even when the manifold and carburettor have been correctly installed, this does not guarantee that you do not have an air leak, or that a leak will not start later on. I always check for leaks around the base of the carburettor and along the face of the manifold. The best method is to squirt petrol out of an oil can around the manifold face and carburettor with the engine running. If there is a leak, the engine will run rich or may even stop. If you use this method of leak detection, always have a fire extinguisher at hand. Usually there are no problems, but a backfire could start a serious fire with all that fuel running around.

Some types of carburettor have a tendency to loosen the screws that attach the body to the base. These should be regularly tightened, but do not use Loctite on them as this may cause the soft threads to strip.

Many engines loosen the manifold retaining bolts very quickly, so these should be Loctited. If the same bolts also retain the exhaust manifold, the Loctite will not work too well, so you should tighten them frequently. By paying careful attention to this, most induction air leaks can be avoided.

THROTTLE LINKAGE

When you are setting up the carburettors or fuel injection, spend some time sorting out a good throttle linkage. This is not usually a problem when a single carburettor is used, but with a multiple-unit set-up it is essential that the butterflies open together and that they remain synchronised right through to the full throttle position. If any of the throttle linkages are cranked, it is possible for the butterflies to break open simultaneously, but as full throttle is reached the throttle plates could be opened to differing angles.

While you are attending to the throttle linkage, you should also see that you are indeed getting full throttle. Ask someone to hold the pedal flat to the floor while you check that the butterflies are opening fully (be sure that the engine is not running!

...el bowls and upse... ...has also been nic...
...ain when delicate...

FUEL PUMP AND FUEL LINE SELECTION

Earlier in this chapter we had a l... at the fu... flow requirements of modified en... As well as considering the flow through the ... we must also check th... the fuel pump and fuel lines will flow the ...quired quantity of fuel so that the fue... ...mains full at sustained high rpm. If the bowl empties slightly, this wil... immediately lean the fuel/air mixture, reducing power and possibly causing damag... pistons and valves.

Most fuel pumps are flow rated in gallons per hour (gph). At first glance ... seem very easy to match a given pump to a particular engine – if the engine ... gph of petrol at maximum power rpm, hook up a pump rated ...

Unfortunately there is a lot more involved than this. First, the output volume ... pump will drop off as you increase the restriction in the fuel lines and need... Many pumps are rated 'free flow' without any inlet or outlet restrictio... ...a Holley 110gph electric pump will flow not 110gph but something closer ... when used with 3... fuel ...

...ver a liquid flows through a closed pipe there is bound to be a cert... ...ount of restriction due to friction in the pipe, sharp corners and the needle and sea... This causes a drop in pressure from one end of the pipe to the other. The pump ma... ...put pressure of 5psi, but at the other end of the pipe the pressure may ... ...st due to restriction.

It does not matter whether you have the fuel pump mounted on the engine, drawing fuel from the tank at the rear of the car, or whether the pump is mounted near the tank at the back of the car, pushing fuel up to the engine. The pressure drop due to line restrictions is the same in both cases.

The pressure drop increases as the square of the volume of flow through the pipe, so if you double the gph flow the pressure loss would be multiplied four times. Conversely, the pressure drop is inversely proportional to the square of the internal diameter of the pipe. Thus if you were to use a pipe twice the diameter, a given gph flow rate would only have one-quarter the pressure loss. Obviously when large increases in an engine's fuel requirements have been made, larger-diameter fuel lines are in order.

A car's acceleration also effectively causes a flow restriction by tending to pull the fuel back to the tank; this can account for a 2–3psi pressure loss in a drag car.

In view of these factors, the fuel pump selected should be able to produce more than the gph flow requirement of the engine. I use a rule of thumb for a petrol-fired racing engine of 1.5 times the theoretical fuel flow requirement, or 0.08 times the maximum horsepower. Because a high-performance road engine is not usually held at sustained high rpm, the required flow is considerably less.

When selecting a pump, be sure that you know whether it is rated in Imperial or US gallons. If you live outside the US you will run into trouble if you calculate your fuel flow to be, say, 70gph and you fit a 70gph American pump. One US gallon is 0.8 of an Imperial gallon, so the American pump will actually flow 56 Imperial gph. (The above rule of thumb calculates the fuel flow requirement in Imperial gallons per hour when working from the maximum horsepower.)

Most high-flow American fuel pumps have a very high delivery pressure, which may cause English and European carburettors to flood. Holley carburettors are designed to operate at an idle speed pressure of 6–7psi, and 4.5psi at maximum speed. Weber carburettors should not have a delivery pressure (at the carburettor) of more than 4–4½psi at idle, and the pressure at maximum speed should not be less than 3psi.

To regulate the pressure, you will have to use an in-line pressure regulator or a bypass line. Holley make a regulator for use with their high-pressure pumps, which pump at up to 15psi unregulated. The regulator must be fitted close to the carburettor, then adjusted according to the fuel pressure gauge reading at idle.

Some racers prefer to use a bypass line from the pump outlet back to the tank, to reduce the pressure. This system will work very well, but a lot of time can be spent finding what size of restriction plug should be fitted in the bypass line to get the required fuel pressure. I would recommend that you use a ¼in line and start your testing using a restrictor with a 1/16in hole.

The front-mounted fuel pump is acceptable in most instances for engines in up to medium semi-race tune. Above this degree of modification an electric pump mounted near the fuel tank should be used.

The main weakness of using a front-mounted pump is that it aggravates vapour lock. A high performance engine requires a lot of fuel so the line from the pump back to the tank will always be subjected to suction or a vacuum. This means that the fuel in the pipe is going to boil at a much lower temperature than it would if at normal atmospheric pressure. The only way around this is to use a rear mounted electric pump.

In large displacement racing machines it is at times necessary to use two or three electric pumps. These must be connected in parallel, so that each has an inlet line running to the fuel tank. The outlet lines may be left separate until they reach the front of the car, or they may be connected to a single large-bore fuel line.

When installing an electric pump I recommend that a Holley safety switch be included in the electrical circuit, so that the pump will not work unless there is oil [illegible] of fuel could otherwise be pumped over everything before the pump was switched off, if an ordinary switch was used. With the ever-present risk of fire, this would be extremely dangerous.

The fuel line must be protected from possible mechanical damage for the same reason. Flying stones or abrasions between rubbing parts could puncture the line and allow fuel spillage. To avoid this, the line must be carefully routed, and sections exposed to stone damage must be shielded with suitable covering.

The line may also fracture because of vibration. To overcome this, a flexible [illegible] and also the fuel tank. If an electric pump is used, flexible line must be used on the inlet and outlet as these pumps are subject to considerable vibration.

When routeing the line, be sure to keep it well away from the engine and the [illegible] into vapour upon passing into the fuel bowl, creating an under-bonnet fire hazard and upsetting the carburettor fuel metering.

A fuel filter should be fitted in the system to remove dirt and water from the fuel. In racing applications it may be advisable to use two filters in parallel, to reduce flow restriction. A filter should never be fitted on the suction side of the fuel pump.

WEBER CARBURETTOR

Weber carburettors are known the world over as about the best that money can buy. However, many people do not realise that there are Webers and Webers. Some are simply metering devices for use on baby Fiats, others, such as the DCOE and IDA series, are racing carburettors that can be tuned to work very well, even on mildly modified street engines.

Many people feel that the Weber is difficult to tune, they never seem to be able to get it to work correctly. I would say that the Weber is one of the very easiest of [illegible] to tune, and that it holds its tune even when subjected to severe banging on a rally car.

The Weber DCOE is a sidedraught unit available with throttle bore diameters of 40, 42, 45, 48 and 50mm. Because of the large range of venturis (commonly called 'chokes') available, these five basic carburettors can be tuned to suit any engine.

The downdraught Weber IDA is available in sizes of 40, 44, 46 and 48mm. It appears to be a DCOE turned up in the air, but really there is little similarity between the two types.

Table 4.7 indicates the carburettor size and choke size recommended for cylinders of various displacements. There will be exceptions to these recommendations but generally you will find them to be very close.

Table 4.7a Weber carburettor and choke size for a four-cylinder engine with siamesed inlet ports

Cylinder displacement (cc)	Choke size (mm)				Recommended carburettor
			Full-race		
	Sports	*Semi-race*	*7,000rpm*	*8,000rpm*	
200	27	28	30	31	40 DCOE
250		31	32	34	42 DCOE
	28	32	33	35	40 DCOE
325	33	35			42 DCOE
	32	34	36	38	48 IDA or 45 DCOE
400	33	35	37	40	48 IDA or 45 DCOE
450	34	36	38	40	48 IDA or 45 DCOE
500	36	38	40		45 DCOE
			40	42	48 IDA or 48 DCOE

Table 4.7b Weber carburettor and choke size for a four-cylinder engine with an inlet port per cylinder

Cylinder displacement (cc)	Choke size (mm)					Recommended carburettor
			Full-race			
	Sports	*Semi-race*	*7,500rpm*	*8,500rpm*	*9,500rpm*	
200				29	32	2 x 40 DCOE
250	27	28	29	32	34	2 x 40 DCOE
325	29	32	34	36		2 x 40 DCOE
				35	38	2 x 45 DCOE
400	32	34	36			2 x 40 IDA or 2 x 40 DCOE
			36	40	40	2 x 44 IDA or 2 x 45 DCOE
				40	42	2 x 48 DCOE
450	33	35	36			2 x 40 IDA or 2 x 40 DCOE
			37	40		2 x 45 DCOE
				40	42	2 x 48 DCOE
500	34	36	38	40		2 x 45 DCOE
			38	42	45	2 x 48 IDA or 2 x 48 DCOE
				42	45	2 x 50 DCOE
550	35	37	40			2 x 45 DCOE
	32	35	39	42	45	2 x 48 IDA or 2 x 48 DCOE
				42	45	2 x 50 DCOE
600	34	36	40	43		2 x 48 IDA or 2 x 48 DCOE

Table 4.7c Weber carburettor and choke size for a six-cylinder engine with an inlet port per cylinder

Cylinder displacement (cc)	Choke size (mm)				Recommended carburettor
			Full-race		
	Sports	*Semi-race*	*6,500rpm*	*7,500rpm*	
330	27	29	30	32	3 x 40 DCOF
400	28	30	33	36	3 x 40 DCOE
450	30	31	34	37	3 x 45 DCOE
500	31	32	36	40	3 x 45 DCOE
				38	3 x 48 DCOF
600	33	34	38	40	3 x 45 DCOF
			37	40	3 x 48 DCOF
700	36	38	40		3 x 45 DCOF
	35	37	39	42	3 x 48 DCOF
800	36	38	42		3 x 48 DCOF

Table 4.7d Weber carburettor and choke [illegible] V8 engine with an inlet port per cylinder

Cylinder displacement (cc)	Choke size (mm)					Recommended carburettor
				Full race		
	Sports	*Semi race*	*7,000rpm*	*8,000rpm*	*9,000rpm*	
500			38	39	42	4 x 48 DCOE or 4 x 48 IDA
	36	38	42	44		2 x 48 DCOE
600		36	38	40	43	4 x 48 DCOE or 4 x 48 IDA
	36	38	42	44		2 x 48 DCOE
700	36	38	42	44		4 x 48 DCOE or 4 x 48 IDA
	38	40	43	45		2 x 48 DCOE
800	38	42	44	45		4 x 48 DCOE or 4 x 48 IDA
	38	42	45			2 x 48 DCOE
900	40	42	45			4 x 48 DCOE or 4 x 48 IDA
	40	42	45			2 x 48 DCOE
1000	42	42	45			4 x 48 DCOE or 4 x 48 IDA
	40	42	45			2 x 48 DCOE

Note: the twin 48 DCOE set up uses a Waneford cross-over manifold mounted on a single or two-plane manifold

The auxiliary (also called the secondary or booster) venturi is located in the throat of the carburettor, in front of the main venturi (choke). The number stamped on this item indicates the size of the fuel spray hole that connects to the main fuel circuit. Usually, a 4.5mm auxiliary venturi is the correct size, but at times a 3.5 or 5mm may be needed. The influence of the flow passage size is felt more markedly at high rpm

Engine	[illegible]	Idle	Auxiliary venturi
[illegible]	[illegible]	50F11	4.5
[illegible]	[illegible]	50F11	4.5
[illegible]	[illegible]	50F8	4.5
[illegible]	[illegible]	45F9	5.0
[illegible]	[illegible]	45F6	5.0
[illegible]	[illegible]	50F9	3.5
[illegible]	[illegible]	50F9	3.5
[illegible]	[illegible]	50F9	3.5
Austin Healey	[illegible]	120 & 60F10	4.5
	[illegible]	50F2	3.5
Avenger	[illegible]	50F2	4.5
	[illegible]	45F8	4.5
Aston Martin T C	[illegible]	45F8	4.5
BMW ohc	[illegible]	50F6	4.5
Chrysler Slant	[illegible]	45F8	5.0
	[illegible]	50F8	3.5
Chrysler Hemi 6	[illegible]	50F8	3.5
[illegible]	[illegible]	55F9	4.5
[illegible]	[illegible]	55F9	4.5
[illegible]	[illegible]	55F8	4.5
[illegible]	[illegible]	60F9	4.5
[illegible]	[illegible]	55F9	4.5
[illegible]	[illegible]	60F8	4.5
[illegible]	[illegible]	100 & 65F10	4.5
[illegible]	[illegible]	70F6	4.5
[illegible]	[illegible]	55F9	4.5
[illegible]	[illegible]	40F9	4.5
[illegible]	[illegible]	45F9	4.5
[illegible]	[illegible]	45F9	4.5
[illegible]	[illegible]	45F9	4.5

	[illegible]	[illegible]	[illegible]	33	[illegible]	[illegible]	[illegible]	40	[illegible]	45F8	4.5
	[illegible]	[illegible]	[illegible]	34	[illegible]	[illegible]	[illegible]	45	[illegible]	50F8	4.5
	[illegible]	[illegible]	[illegible]	34	[illegible]	[illegible]	[illegible]	40	[illegible]	45F8	4.5
[illegible]	[illegible]	[illegible]	[illegible]	42	[illegible]	[illegible]	[illegible]	50	[illegible]	50F9	4.5
[illegible]	[illegible]	[illegible]	[illegible]	34	[illegible]	[illegible]	[illegible]	45	[illegible]	45F9	4.5
	[illegible]	[illegible]	[illegible]	37	[illegible]	[illegible]	[illegible]	40	[illegible]	45F9	4.5
	[illegible]	[illegible]	[illegible]	38	[illegible]	[illegible]	[illegible]	60	[illegible]	60F9	4.5
	[illegible]	[illegible]	[illegible]	42	[illegible]	[illegible]	[illegible]	60	[illegible]	60F9	4.5
	[illegible]	[illegible]	[illegible]	43	[illegible]	[illegible]	[illegible]	55	[illegible]	120 & 55F10	4.5
	[illegible]	[illegible]	[illegible]	36		[illegible]	[illegible]	55	[illegible]	60F8	4.5
	[illegible]	[illegible]	[illegible]	32	[illegible]	[illegible]		35	[illegible]	45F9	4.5
	[illegible]	[illegible]	[illegible]	33		[illegible]		40	[illegible]	45F8	4.5
[illegible]	[illegible]	[illegible]	[illegible]	38		[illegible]		40	[illegible]	65F8	3.5
	[illegible]	[illegible]	1 x 45 DCOE	34		[illegible]	[illegible]	50	[illegible]	50F9	3.5
	[illegible]	[illegible]	1 x 45 DCOE	36		[illegible]		60	[illegible]	50F9	5.0
	[illegible]	[illegible]	2 x 40 DCOE	30		[illegible]		40	[illegible]	45F6	4.5
[illegible]	[illegible]	[illegible]	2 x 40 DCOE	33	[illegible]	[illegible]		40	[illegible]	50F8	4.5
	[illegible]	[illegible]	2 x 45 DCOE	34	[illegible]	[illegible]		40	[illegible]	50F8	4.5
	[illegible]	[illegible]	2 x 48 DCOE	[illegible]		[illegible]		50	[illegible]	55F9	4.5
Porsche ohc	[illegible]	[illegible]	2 x 40 IDA	30		[illegible]		50	[illegible]	55	4.5
ohc	[illegible]	[illegible]	2 x 40 IDA	32		[illegible]		50	[illegible]	55	4.5
ohc	[illegible]	[illegible]	2 x 46 IDA	42		[illegible]		50	[illegible]	70	4.5
Renault x	[illegible]	[illegible]	2 x 40 DCOE	32		[illegible]	F15	35	[illegible]	45F8	4.5
x	[illegible]	[illegible]	2 x 40 DCOE	34		[illegible]	F15	40	[illegible]	45F8	4.5
Triumph	[illegible]	[illegible]	3 x 45 DCOE	33	[illegible]		F15	40	[illegible]	45F9	4.5
	[illegible]	[illegible]	2 x 45 DCOE	34	[illegible]		F15	50	[illegible]	50F8	4.5
	[illegible]	[illegible]	2 x 48 DCOE	[illegible]			F15	50	[illegible]	50F9	4.5
[illegible]	[illegible]	[illegible]	[illegible]	35			F15	50	[illegible]	50F8	4.5
		[illegible]	2 x 48 IDA	40		[illegible]	F2	50	[illegible]	120 & 55F10	4.5
		[illegible]	2 x 44 IDA	35		[illegible]	F11	45	[illegible]	100 & 50F10	4.5
		[illegible]	2 x 40 IDA	32		[illegible]	F11	40	[illegible]	100 & 50F10	4.5

Note. Rally tune indicates an engine tuned for club tarmac rallies
T/C = Twin-cam, T/C 4V = Twin-cam four valve, 4V = ohc four-valve, x = push rod crossflow

WEBER MAIN JET

In Weber carburettors the main jet, emulsion tube and air corrector jet are pressed together to form [illegible] of pliers, but never grip the emulsion tube with pliers as it may be damaged.

The main jet is pressed into the bottom of the emulsion tube. It is stamped with a number indicating its nominal bore diameter (eg a 175 jet has a 1.75mm bore).

As a starting point, the main jet size can be selected by multiplying the size of the main venturi by 3.9 to 4.3 (eg a 32mm choke will usually have a main jet size of 125 to 135). [illegible] way off the mark. If you check Table 4.8 you will see that this is usually in the case of racing engines with siamesed inlet ports. In this instance the multiplying factor is 4.6 to 5.0.

The air corrector (or air bleed) jet is pressed into the top of the emulsion tube. Air corrector jets from 1.50 to 2.00mm are more commonly used. By increasing the diameter of this [illegible] A change in the size of the main jet changes the mixture strength uniformly at both high and low rpm.

With DCOE carburettors, the air corrector jet will usually be 0.30 to 0.50mm larger than the main jet size, eg with a 125 main jet try a 155 to 175 air corrector. In general, road engines will use the larger jets and racing engines the smaller jets.

Engines using IDA carburettors will generally require an air corrector jet around [illegible] to 0.50mm, smaller than the main jet.

The emulsion tube emulsifies the fuel previously metered by the main jet. It affects engine performance more at small throttle opening angles and during acceleration.

The fuel well into which the emulsion tube fits in the carburettor is of a fixed [illegible] of holes, but the tubes are of a different diameter.

The amount of fuel available to be drawn into the air stream is governed by the diameter of the emulsion tube [illegible] has such an effect on throttle response and mid-range running.

Once we have selected main and air corrector jets to meet the engine's high rpm fuel requirements, we have to set about finding an emulsion tube of the correct [illegible] is thicker, with holes all the way down. For most applications F2, F15 and F16 tubes are used; F11, F9 and F7 tubes are used in a smaller number of applications.

For the majority of engines with DCOE carburettors using chokes smaller than [illegible] four-cylinder engines using one DCOE carburettor generally use an F2 tube, however, some may need an F9. When IDA carburettors are fitted, an F2 or F7 will be correct in the majority of applications.

WEBER ACCELERATOR PUMP JET

As in every carburettor, the accelerator pump circuit in the Weber supplies a shot of fuel to assist in smooth

power circuit to supply additional fuel at high speeds

fuel/air mixture for high-speed operation.

jet you will need a special screwdriver with a 'screw grip' attachment.

The length of the pump rod governs the amount of fuel in the pump well, the

shortened by the use of a collar

The promptness of fuel delivery is controlled partly by the bleed jet and partly by

length of the pump rod or the strength

vantageous to use a strong

the 1300 Mini Cooper 'S'

WEBER IDLE JET

Both the DCOE and IDA model carburettors have an idle jet assembly that meters fuel

In DCOE carburettors the idle jet has a fuel hole and an air bleed hole. The fue

consideration when working out the table.

All IDA idle jets are coded F10. The F10 jet does not have an air bleed hole.

uses a 60 F10 idle jet and a 120 jet holder. This indicates that the jet holder has a

larger air bleed hole leans the idle mixture.

Table 4.9 Weber idle jet air bleed characteristics

Air bleed code	
F6	
F12	
F9	
F8	
F11	richer
F14	
F13	
F1	
F3	

Note F10 idle jet has no air bleed hole and is for use in IDA carburettors only

Table 4.10 Weber idle jet fuel hole selection

Cylinder displacement (cc)	Fuel jet size
200	35
250	40
325	45
400	45–50
450	50
500	50
550	55–60
600	55–60
700	65–70
800	70–75
900	75
1000	80

Note where two cylinders share one carburettor barrel, or where the engine has siamesed inlet ports, the idle jet size should be increased one size larger than indicated Engines with wild camshafts may require jets two sizes larger due to poor fuel distribution at low rpm when two cylinders share one carburettor barrel

To determine the approximately correct idle jet refer to Table 4.10, which sets out the sizes of the fuel metering holes If, for example, the cylinder capacity is 400cc, you will require a 50 F8 or 50 F9 jet with which to begin testing I have arbitrarily selected an F8 or F9 air bleed hole in this instance as either size is correct in 80% of cases; the other 20% use an F2 or F6 air bleed

If you were using an IDA carburettor you would select a 50 F10 idle jet in the above example, and you would use either a 100 or 120 jet holder

After having made your idle jet selection [illegible]

[illegible] smoothest idle Remember that if the correct idle jets have been selected, the mixture screws should be 1/4–1 1/4 turns open

[illegible] analysers read the other way around to give the air/fuel ratio, ie 11 5 or 12.5:1)

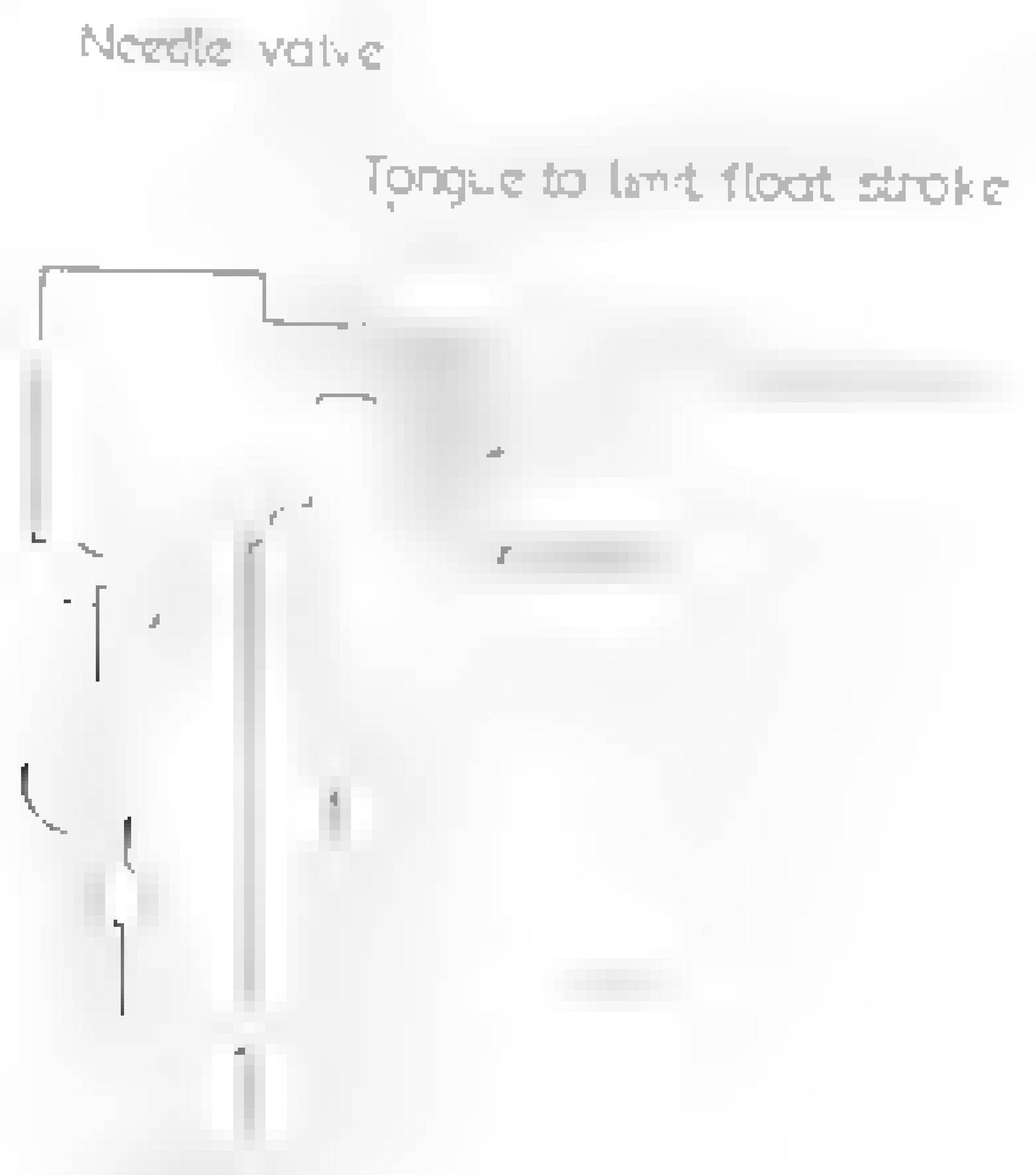
The [illegible] necessary to use the choke to get a Weberised engine started, even when the car has [illegible] is all that is required

Usually, you have more trouble starting a Weber-equipped engine when it is hot, [illegible] you will end up with a real problem

DCOE FLOAT LEVEL ADJUSTMENT

The float level adjustment is critical on all carburettors and especially so on [illegible] have an additional adjustment to ensure that the float drops enough to open the needle valve completely when fuel demand is high

Figure 4 11 Weber DCOE float levelling



To adjust the float level and float stroke on the DCOE it is necessary to remove the top of the carburettor. The jet inspection cover also has to be removed before the top becomes free. Take care not to damage the floats as they are very delicate, being made of sheet 0.16–0.20mm thick; do not blow them with compressed air.

With the top removed, ensure that the needle valve is tightly screwed into its housing and check that the spring-loaded ball incorporated in the valve is not jammed.

When checking the float level, hold the carburettor top in the vertical position (as shown in Figure 4.11) and, with the float tongue just contacting the spring-loaded [illegible] (measurement A). If the float level is not correct, push out the float pivot pin and remove the float, then carefully bend the float tongue to obtain the correct dimension.

After the float level has been checked and adjusted, hold the float fully open and measure the float stroke (measurement B). If the stroke requires adjustment, bend the tongue to increase or decrease the stroke.

To measure the distances A and B, Weber make a special float gauge, but as these are often difficult to obtain I have made my own set of gauges out of steel or aluminium bar. The gauge should be about 8 inches long, dimension A thick and dimension B wide. Be sure to machine four grooves in the bar (two in each of two faces) so that the gauge clears the soldered joining band of the float halves. Without these clearance grooves you will not be able to adjust the float accurately as you will be measuring between the soldered join and the carburettor top gasket, rather than between the floats proper and the gasket.

The actual measurements A and B will vary from one application to another, depending primarily on the angle at which the carburettor is mounted. Weber DCOE carburettors are designed to be mounted at 0–5° from horizontal, and usually at that angle dimension A is 8.5mm and dimension B 15mm.

In some applications the manifold manufacturer finds it necessary to change the mounting angle, so that the carburettor will clear some part of the body or exhaust system. When this is done, a special float level setting is required. The manifold manufacturer should tell you what setting is required when you buy his manifold.

The majority of Austin Mini manifolds have the Weber mounted at an angle of 7–7½°; in this instance dimension A is 7mm and dimension B 15mm. Some cars (eg Aston Martin DB4, Alfa Romeo 2600) use a high float level with dimension A 5mm and B 13.5mm.

IDA FLOAT LEVEL ADJUSTMENT

To check the float level of IDA model carburettors you will need to make a set of gauges or use Weber gauges Nos 9620.175.1839, 9620.175.1840 and 9620.175.2411 and spring No 9620.175.1329.

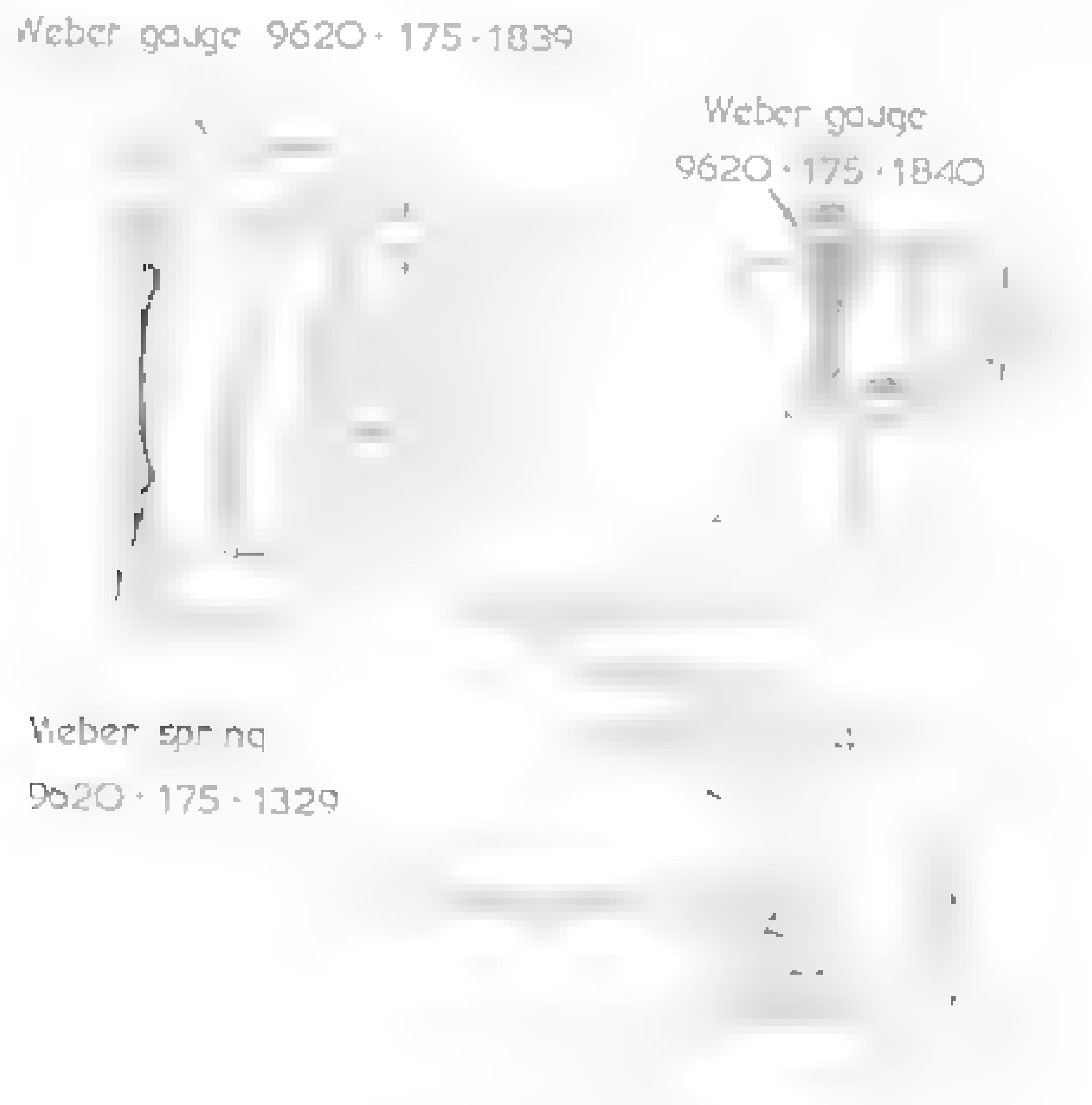
Referring to Figure 4.12, the check is carried out as follows.

- Remove the carburettor top and the gasket and insert the spring (No 9620.175.1329) between the float and the side of the bowl.
- Position gauge No 9620.175.1840 on the carburettor body and manipulate the float so that the tongue just contacts the tip of the gauge. (Note that the gauge must first be adjusted to 24.2mm as shown.)

- With the float in this poss.tion cl eck using gauge No 9620 175.1839 that the float is 5.5–6mm above the carburettor body Ensure that the gasket is removed when this is being checked Should the level be incorrect, bend the float tongue to obtain the correct level
- Finally, using gauge No 9620.175 2411 check that the needle valve ball is 25mm from the surface of the [illegible] be inverted and the needle valve ball must not be depressed

Cars equipped with three-barrel IDA, IDAP, IDL, IDS, IDT and IDTP model carburettors have a float level [illegible] float level gauge No 9620.175.3071 is used, and in this case the float level is 14–14.5mm. All of these three-barrel carburettors have a needle-ball to-carburettor top dimension of 18mm.

Figure 3.12 Weber IDA float levelling



HOW TO MOUNT ON INLET MANIFOLD

There are just a few points to keep in mind when Weber carburettors are fitted. First, I would suggest that you obtain a copy of the *Weber Technical Introduction* 2nd edition (or later) and follow very carefully their installation instructions. Pay careful attention to what they have to say about throttle linkage arrangements so that all the butterflies open together.

DCOE carburettors must never be mounted solidly, but must always have a neoprene insulator block or a light alloy plate with an 'O' ring in both faces fitted between the carburettor and the manifold. This is necessary to reduce carburettor vibration, which causes fuel frothing, flooding and inaccurate metering

When tightening the carburettor mounting bolts, ensure that they are tensioned just enough to effect a good seal, but not enough to squash the 'O' rings or insulator block. Self-locking nuts and, if possible, light double-wound spring washers should be used, as fitted on the Twin Cam Ford Escort, these are designed to be tightened until there is a gap of 0.040in between the coils

Various screws on some model Webers are safety wired, and those parts must always be re-wired whenever the wire is removed or broken. The wire is there for a very good reason, for if a screw were to drop out, the engine would stop or run badly For example, the IDA has the float pivot pin wired; if the pin falls out, the engine floods. The 45 and 48 DCOE have safety-wired grub screws that hold the auxiliary venturis in place. If a screw comes out, the auxiliary venturi can rotate and cut off fuel to the discharge nozzle

TUNING WEBER CARBURETTORS

Before attempting to tune and synchronise the carburettors, the ignition timing, spark plugs, points and float level, etc, should have been adjusted. Check that the carburettor choke control levers are all back against their stops, then bring the engine up to normal operating temperature and switch off. Carefully screw in each idle mixture adjusting screw until it just contacts its seat, then unscrew it three-quarters of a turn. (If you find that you need more or less than three-quarters of a turn to get your engine running, adjust all the mixture screws by the same amount.)

Synchronising the throttle plates in the majority of twin DCOE applications is very easy when an interconnector linkage, as fitted to the Twin Cam Lotus Fords, four-cylinder Alfa Romeos, Renault Gordinis, etc, is used. In these instances start the engine and turn in the idle speed screw until the idle speed reaches 1200rpm (it may be higher if a lumpy cam is used). For the actual synchronisation of the throttle plates you will need either a carburettor synchroniser or a length of quarter inch rubber hose and a good ear

Press the synchroniser over one throat of the carburettor (check the carburettor with the idle speed adjusting screw first), then turn the synchroniser ring so that the indicator float dwells at mid-height in the indicating column. Move the synchroniser to the next carburettor and turn the interconnector linkage screw in or out so that the float again dwells at mid-height

The second carburettor can also be synchronised with the first by listening to the intensity of the hiss of the first carburettor, then adjusting the other to produce a hiss of equal intensity.

After synchronisation, screw out the idle speed screw slightly to reduce the idle speed to 800–1,000rpm

At this point it may be necessary to slightly adjust each mixture screw to obtain the smoothest idle. It should not be necessary to adjust any screw more than one-eighth of a turn, either in or out

[illegible] similar tuning procedure to that outlined is used However, in this instance the throttle control rods should generally all be disconnected and each carburettor must be synchronised to the first carburettor, using the idle speed adjustment screw fitted to [illegible] throttle lever. To complete the job it may be necessary to adjust each mixture screw to smooth out the idle

Some Weber models incorporate an adjustable idle air bleed passage to equalise the idle air flow in every barrel. These carburettors can be identified by the idle air bleed screw with lock nut located on the opposite side of the barrel from the idle [illegible] some DCOE models

Before accurate synchronisation of these models is possible, the air bleed lock nuts must be slackened and the air bleed screws very carefully screwed in until seated Be careful not to use too much pressure or the seats could be damaged

With the air bleeds closed, the butterflies of the carburettors can be synchronised as outlined earlier, using a carburettor synchroniser When this is being carried out ensure that the idle speed is 1,200–1,500rpm Once the throttle plates are synchronised, the idle speed should be reduced to normal (800–1,000rpm). Then use the carburettor [illegible] and adjust the air bleed screws on all the barrels, by screwing them out, to cause the float to rise to the same level While this is being done, it will usually be necessary to lower the idle speed, as the engine will tend to speed up. When the float rises to the same level on every barrel, the air compensation is correct and the lock nuts should be tightened

After the carburettors have been synchronised, it will be necessary to determine if the main and air corrector jets, and the pump jets, are correct. This check is best done on the dyno, but follow up with a road or track test to confirm that the correct jets have been chosen It is not possible to accurately determine the accelerator pump metering on the dyno, so this will have to be checked on the road

When the most suitable jets are fitted, there should be no hesitation during acceleration or when the throttle is suddenly flattened, nor should there be any black smoke (indicating richness) from the exhaust. After a run at sustained high rpm, the spark plugs should be checked against the indications in Table 3.4

SU CARBURETTORS

The SU constant depression carburettor has been around for many years and it still has something to offer those interested in better performance It is a very simple carburettor, but one that can be exceedingly difficult to tune with precision

The majority of SU carburettors are type H and type HS. A few of the type HD 'diaphragm' carburettors are still around but these are used only in 1¾ and 2in sizes. The type H and HD can be identified by having a solidly mounted float chamber, while the HS has an external nylon tube connecting the float chamber to the jet. Table 4.11 indicates the throttle bore diameters of these models.

Table 4.11 SU carburettor types

Type	Throttle bore (in)
H2, HS2	1¼
H4, HS4	1½
H6, HS6, HD6	1¾
H8, HD8	2

displacements. You will note that the SU must always be connected to at least two cylinders, otherwise its constant vacuum principles of operation are upset.

SU carburettors are supplied with either a 0.090, 0.100 or 0.125in jet. These rarely need to be changed except for methanol, when 0.187 or 0.250 jets may be

The volume of fuel introduced into the air stream is controlled by the taper of the needle; needles are made to suit 0.090, 0.100 and 0.125 jets, and there are approximately 300 different needles listed for 0.090 jets alone. When the 0.1870 or 0.250 alcohol jets are fitted, needles intended for use with 0.125 jets may be used, although at times these will be too rich at lower rpm. In this instance you will have to make your own needles from ⅛in bronze welding wire.

Table 4.13 lists just a few 0.090 jet needles. The numbers refer to the diameter of the needle at 12 or 13 different points along its length, commencing at the shoulder at the top of the needle and then continuing every one-eighth of an inch.

Table 4.12 Recommended SU carburettor sizes

Displacement (cc)	Tune	
Four-cylinder engine	*Sports*	*Semi or full race*
850	2 x 1¼in	2 x 1¼in
1,000	2 x 1¼in	2 x 1½in
1,300	2 x 1¼in	2 x 1½in
1,600	2 x 1½in	2 x 1¾in
1,800	2 x 1¾in	2 x 1¾in
2,000	2 x 1¾in	2 x 2in
2,200 plus	2 x 2in	2 x 2in
Six-cylinder engine	*Sports*	*Semi or full race*
2,000	2 x 1¾in or 3 x 1½in	3 x 1½in
2,500	2 x 2in or 3 x 1½in	3 x 1¾in
3,000	2 x 2in or 3 x 1¾in	3 x 2in
3,500 plus	2 x 2in or 3 x 2in	3 x 2in

Table 4.13 SU carburettor needles

MML	CZ	BC	3	H4
0.089	0.089	0.089	0.089	0.089
0.085	0.085	0.085	0.085	0.085
0.0813	0.0827	0.0815	0.0814	0.081
0.078	0.0806	0.0782	0.0785	0.0778
0.074	0.0785	0.0745	0.0765	0.076
0.0707	0.0745	0.0695	0.0744	0.0741
0.0673	0.0727	0.0647	0.0723	0.072
0.0636	0.071	0.060	0.0703	0.0702
0.060	0.0693	0.0557	0.0683	0.0683
0.0563	0.0675	0.0515	0.0661	0.0663
0.053	0.0657	0.0474	0.064	0.064
0.0495	0.064	0.043	0.063	0.062
0.046	0.0625	0.039	0.062	

to find at what point the needle is rich or lean. Then you will have to consult the complete SU needle tables and find a needle that is richer or leaner at the point where you want it to be.

When going from a weak needle to a richer one, it is better to try one about 0.002 ... 0.001 thicker, unless there are signs of excessive richness.

PISTON SPRING AND DASHPOT OIL

... will result in hesitation or spit back when accelerating, due to a lean mixture condition.

The correct oil level for carburettors with a vent hole in the damper piston is ½in above the top of the hollow carburettor piston rod. Carburettors with a non vented damper cap should be filled to ½in below the carburettor piston rod.

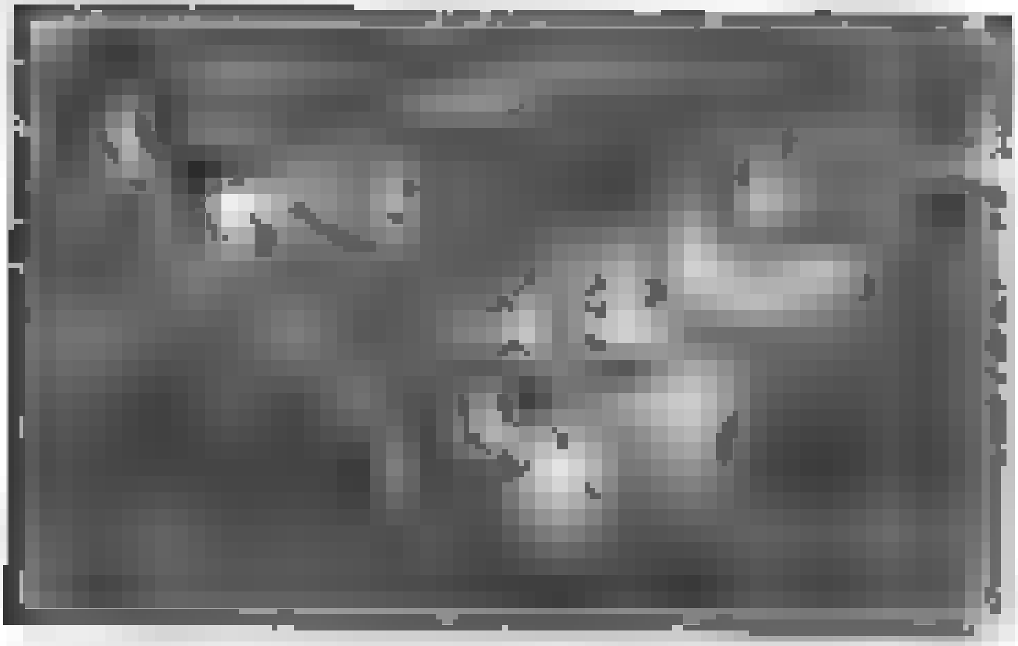
The effect of the oil (and the spring) is to slow down the rate of rise of the ... snap opening of the throttle. This serves the same purpose as an accelerator pump.

Piston springs are identified by a colour code painted on the end coils. The range ... and 12oz green. The correct strength of spring is one that allows the piston to reach its maximum lift at maximum power rpm. This is assuming that carburettors of the correct size are being used.

Table 4.14 SU tuning specification

Engine		Displacement (cc)	Tune	Carb	Needle	Spring
Austin		997	sports	2 x $1\frac{1}{4}$in	GZ	Red
		998	sports	2 x $1\frac{1}{4}$in	GY	Blue
		970	sports	2 x $1\frac{1}{4}$in	AN	Red
		1070	sports	2 x $1\frac{1}{4}$in	H6	Red
		1275	sports	2 x $1\frac{1}{4}$in	M	Red
		970	full-race	2 x $1\frac{1}{2}$in	CP4	Blue
		1070	full-race	2 x $1\frac{1}{2}$in	MME	Blue
		1275	full-race	2 x $1\frac{1}{2}$in	BG	Blue
		1098	sports	2 x $1\frac{1}{2}$in	AM	Blue
		998	semi-race	2 x $1\frac{1}{4}$in	M	Blue
Austin Healey		2912	full-race	3 x 2in	UH	Blue/Black
		2912	sports	3 x 2in	UH	Red/Green
		2912	sports	3 x $1\frac{3}{4}$in	BC	Green
		2912	sports	2 x $1\frac{3}{4}$in	CV	Yellow
Ford		997	sports	2 x $1\frac{1}{4}$in	A5	Blue
		1198	sports	2 x $1\frac{1}{4}$in	H6	Red
		1500	sports	2 x $1\frac{1}{2}$in	CZ	Red
		2553	sports	3 x $1\frac{1}{2}$in	3	Red
		2553	sports	3 x $1\frac{1}{4}$in	ES	Red
		2553	sports	2 x $1\frac{1}{2}$in	7	Yellow
		997	full race	2 x $1\frac{1}{2}$in	AM	Blue
Hillman	ohc	875	sports	2 x $1\frac{1}{4}$in	H4	Blue
		1600	sports	2 x $1\frac{1}{2}$in	QA	Red
Jaguar	T/C	3441	sports	2 x $1\frac{3}{4}$in	TL	Red
	T/C	3781	sports	2 x $1\frac{3}{4}$in	TU	Red
	T/C	3781	sports	3 x 2in	UM	Blue/Black
	T/C	4235	sports	3 x 2in	UM	Blue/Black
	T/C	3441	sports	3 x 2in	UE	Blue/Black
MG	T/C	1588	sports	2 x $1\frac{3}{4}$in	OA6	Red
		1800	sports	2 x $1\frac{1}{2}$in	MB	Red
		1800	full-race	2 x 2in	UVD	Blue/Black
		1098	sports	2 x $1\frac{1}{4}$in	AN	Blue
		1800	semi-race	2 x $1\frac{3}{4}$in	KP	Red
Triumph		1147	sports	2 x $1\frac{1}{4}$in	MO	Red
		1147	full race	2 x $1\frac{1}{2}$in	DB	Blue

Normally a red spring is used, but if the carburettors are the sizes recommended in Table 4.12 for semi- and full-race engines, a blue spring may be required. Conversely, if a pair of carburettors are fitted to a six-cylinder engine, or if carburettors smaller than indicated are used (eg 2 x $1\frac{1}{2}$in on a 2,000cc four-cylinder), a stronger yellow or green spring will probably be required, to allow for a richer acceleration.



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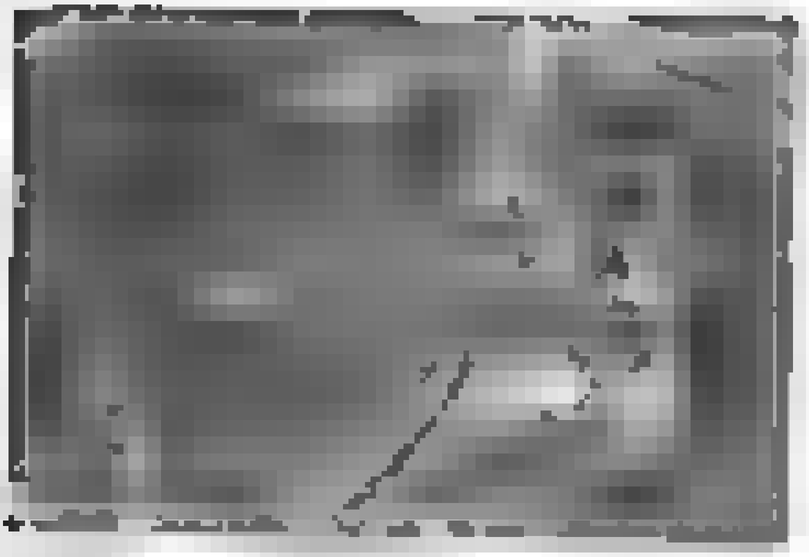

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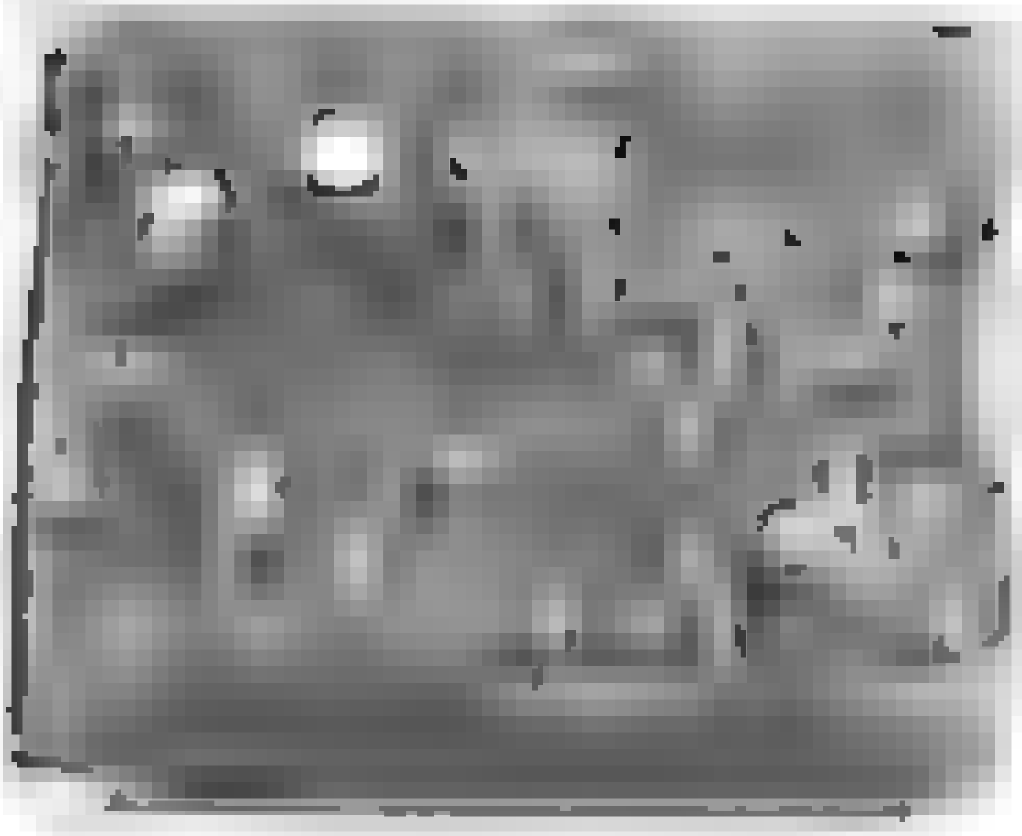
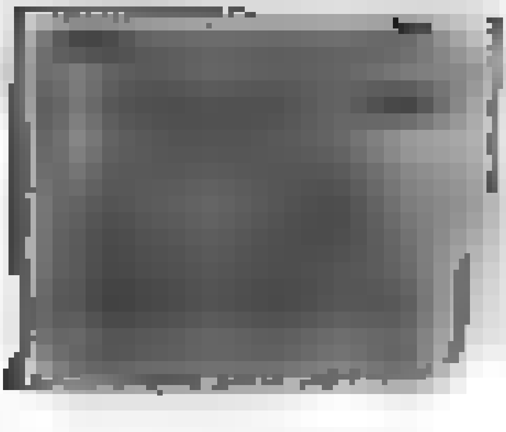
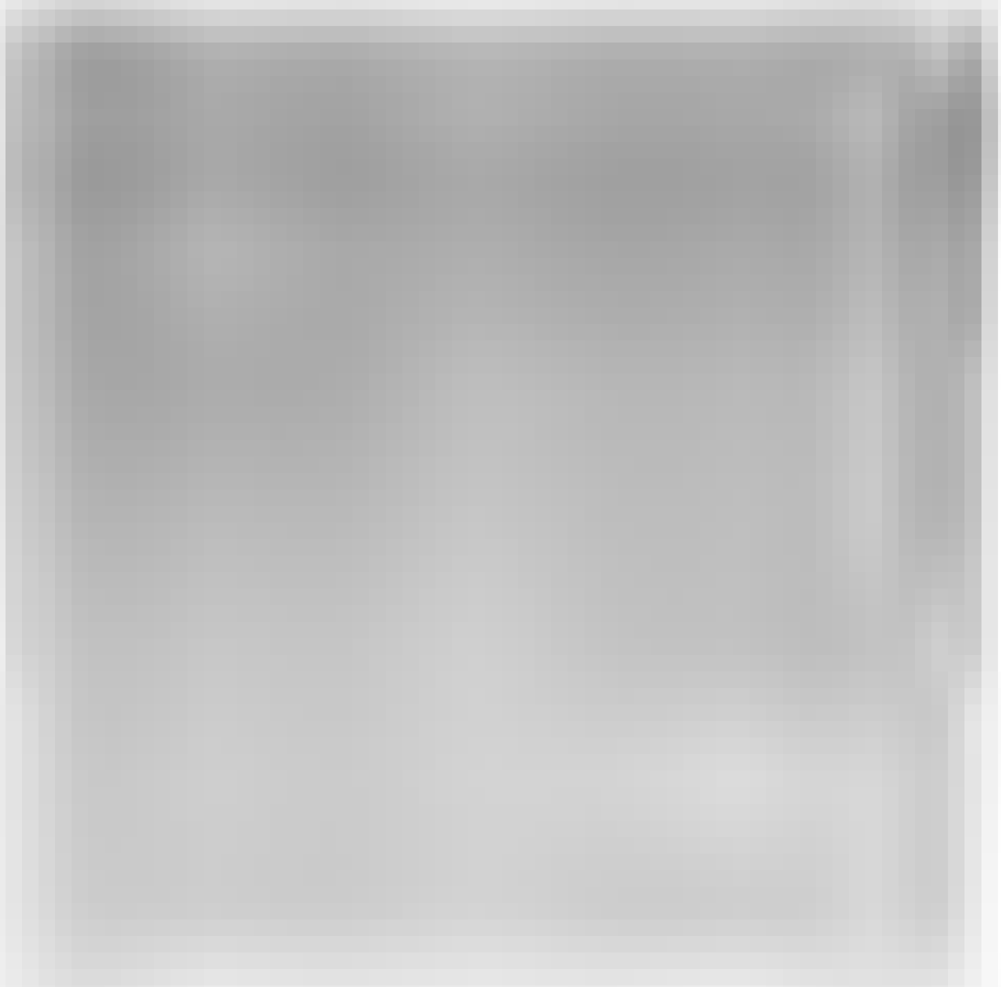
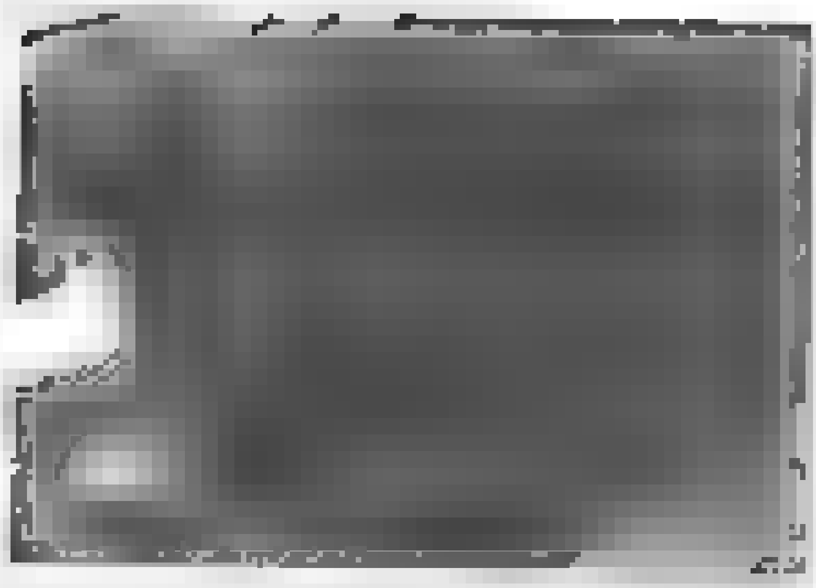
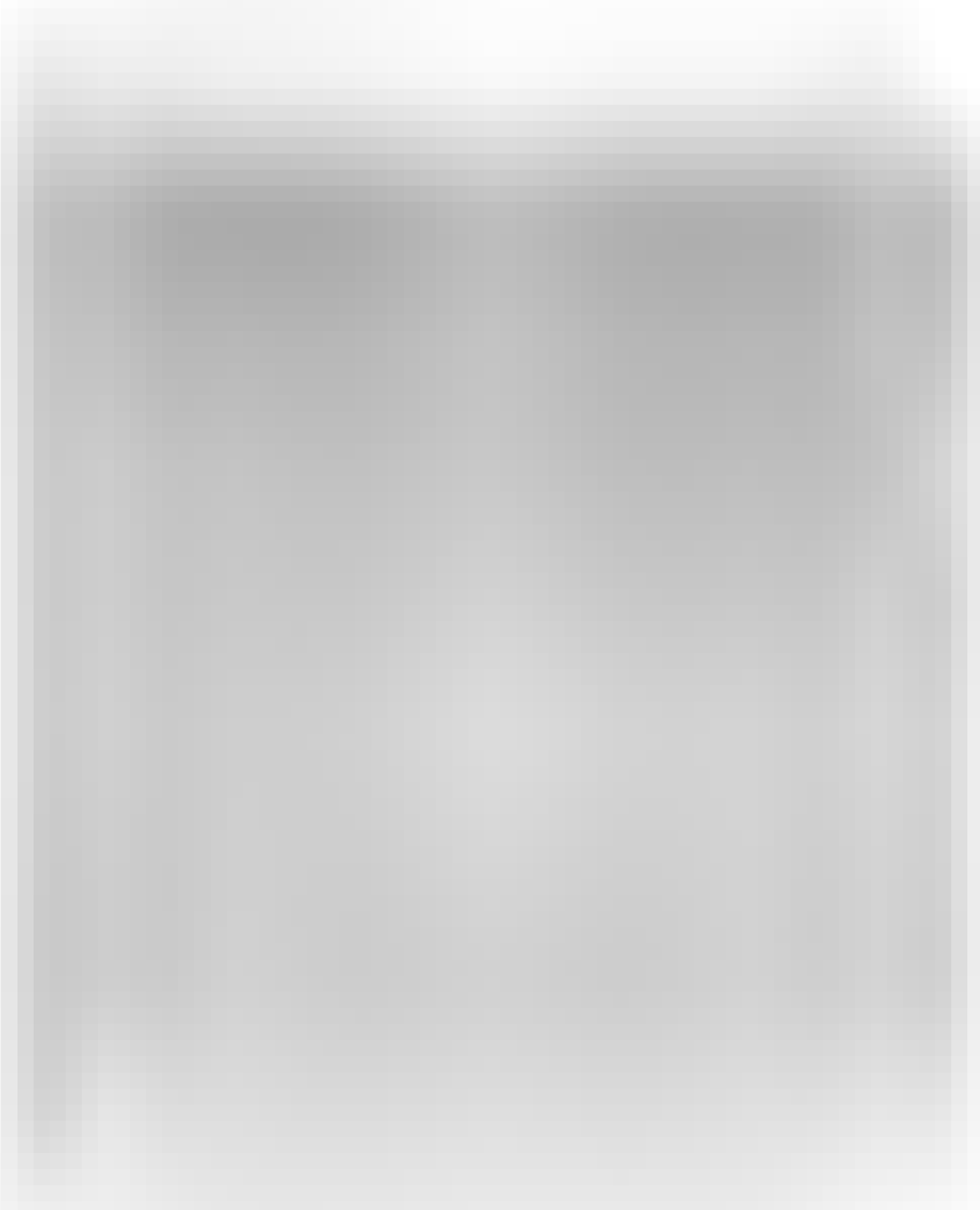
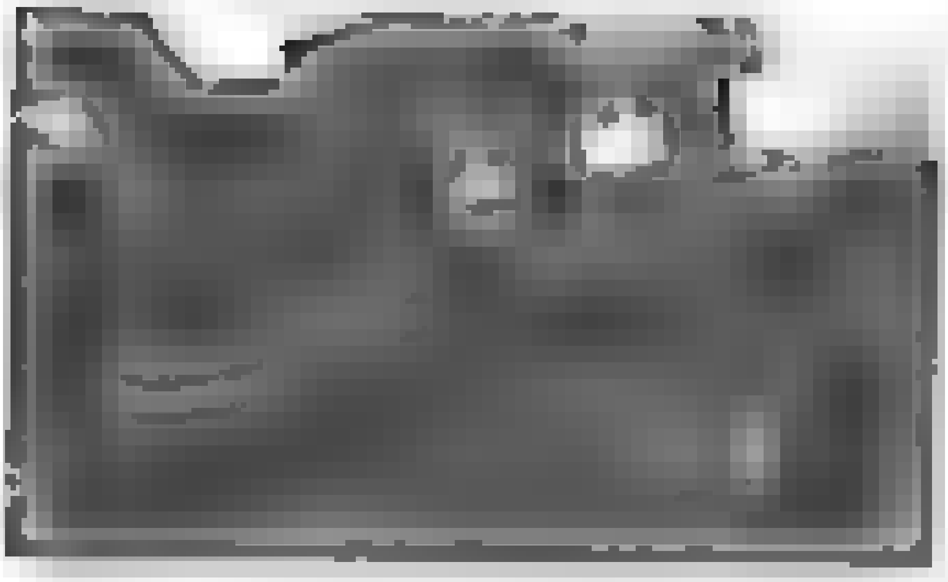


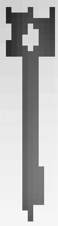


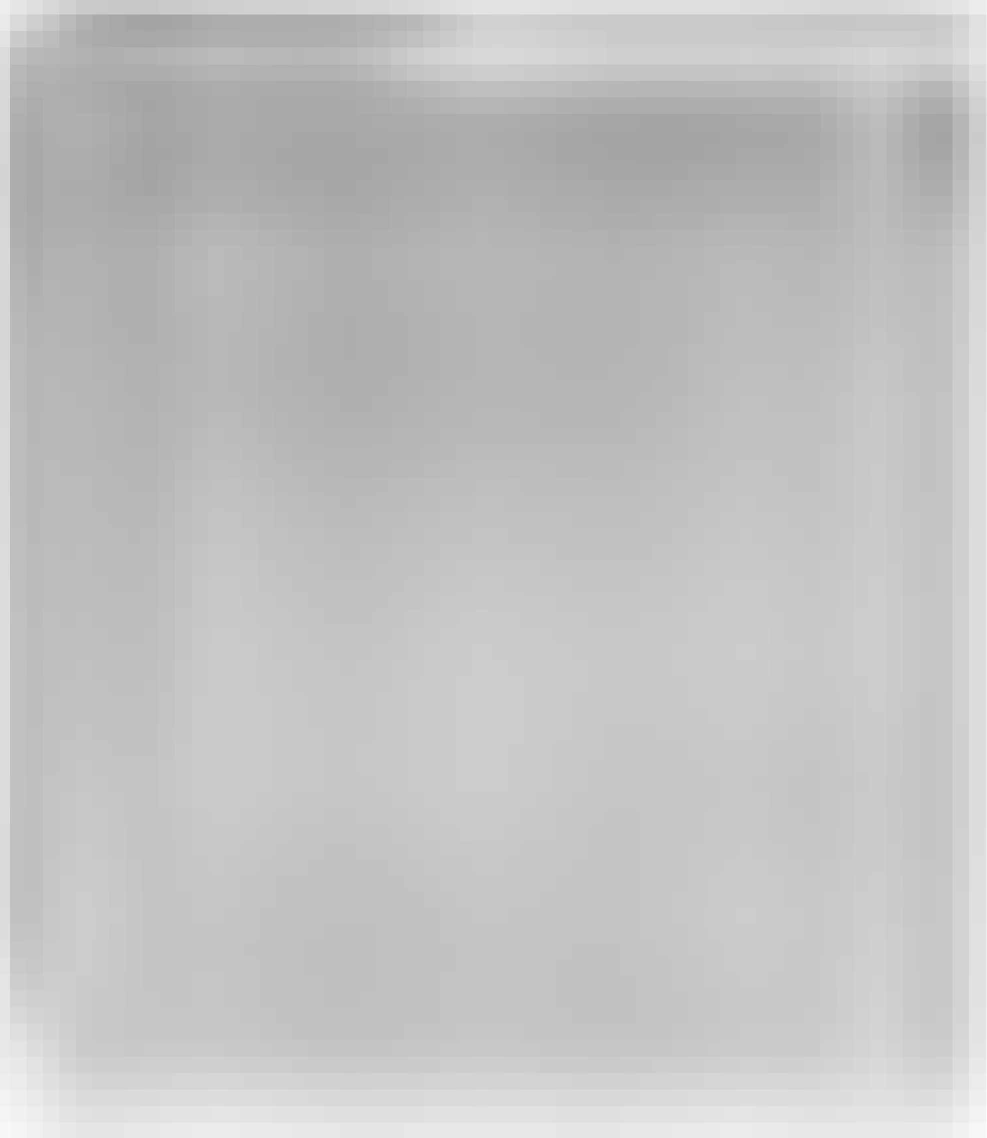


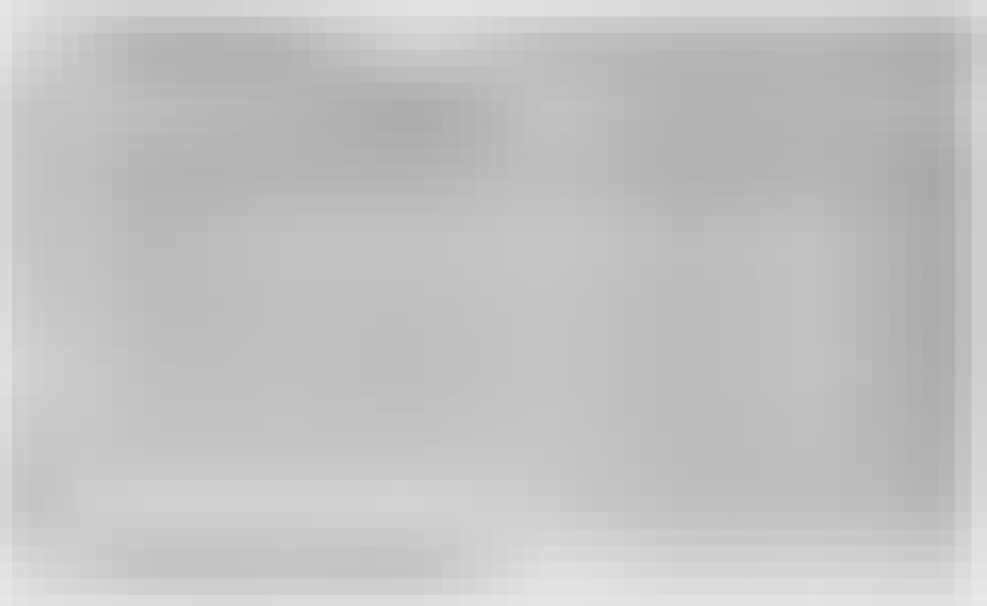












minute. Obviously the flow must not be less than what you have previously calculated

The only problem with this test arises if the car's electrics are a bit wimpish

receiving a voltage boost compared with when the system also has to power the

wiring, good connections and ample alternator capacity

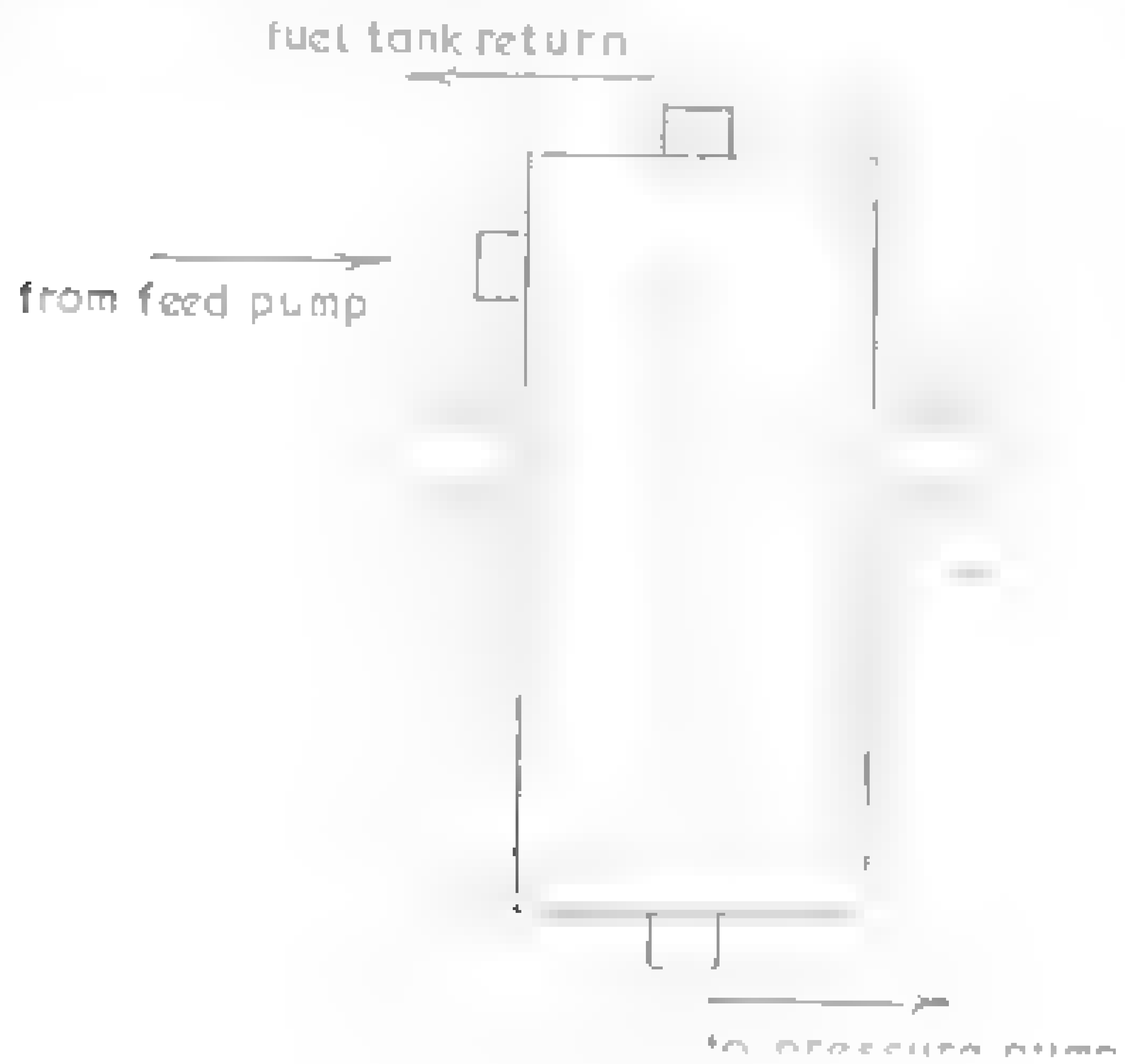
SWIRL POT AND FUEL LINES

against this sort of trouble when converted for competition use

The simplest solution is to run a second fuel pump that pumps into a small-

at least half a minute (Figure 5.15), for our 280hp engine, that means about 1 litre

Figure 5.15 Fuel swirl pot



However, I think it is wiser to reckon on 1ltr per 160–180hp, so I would choose a 1.5–1.8ltr collector for this particular engine. As it is not subjected to pressure it can be fabricated quite easily from 3in tube. Exhaust tubing is acceptable, but if it is to look pretty you might prefer stainless steel or aluminium.

I prefer to use the same type of pump to supply the swirl pot as I would run for the pressure pump. Some, however, choose to use a small pump to feed the swirl pot, and run the fuel return line back to the swirl pot rather than to the fuel tank. I do not believe that this is such a good idea for two reasons: first, unless the swirl pot is fairly large it can run dry at high engine loads and high rpm, and second, at lower engine loadings the fuel will get hot circulating in a closed loop and the engine could become dangerously lean.

Both the suction fuel line from the fuel tank to the pressure pump and the pressure line from the pump to the fuel rail must be of adequate size and with bends of sufficiently generous radii to not unduly restrict fuel flow. Table 5.10 sets out the approximate hp levels that various sizes can support. Note that when stainless steel braided hoses are used these will have an id smaller than that indicated in the table, but as only small lengths are involved this should not be a concern. Incidently, these hoses are identified by a dash number representing 1/16in, thus a '–6' hose is for connecting to a 3/8in (6/16) hardline tube. On the pressure side a hose of this size will flow enough fuel for a 500hp engine.

Table 5.10 Fuel line fuel flow

Tube od	Tube id	hp pressure	hp suction*
1/4	.180	165	110
5/16	.243	300	200
3/8	.319	500	335
1/2	.444	1000	675

* These figures can also be used to estimate fuel line size for engines with carbs

Chapter 6

Fuel and Fuel Boosters

Fuel to a large extent is still a great mystery. Most racers and enthusiasts know a little about petrol (gasoline), methanol and nitromethane. They know that an [illegible] could produce an overall variation of 5% in power outputs, or that one formulation gave significantly sharper throttle response.

WHAT FUEL OCTANE RATINGS MEAN

The problem has been over the years we have been unwittingly led to understand that [illegible] response and power output there will be measurable differences.

Since this confusion exists as to what octane ratings really mean it is best to look [illegible] management controls, or was not retuned in any way to take advantage of the octane increase.

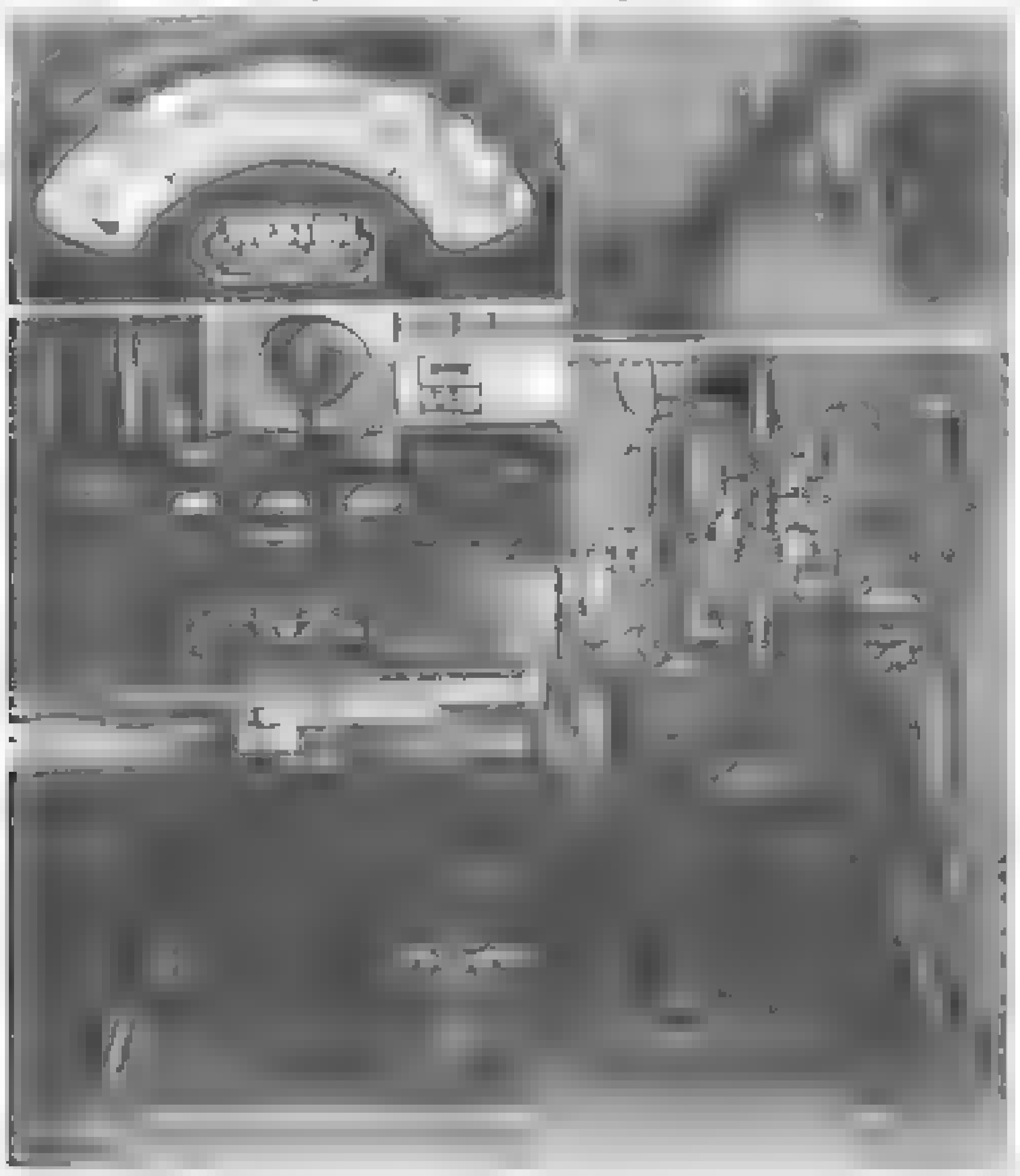
HISTORY OF OCTANE STANDARDS

To better appreciate what octane numbers are all about we have to take a t[illegible] history to the time of World War I. At that time it was found that aircraft engines would suddenly self-destruct through detonation. An engine might run fine on one load of fuel, but punch holes in the pistons on the next batch. The problem for the fuel refiners was the fuels seemed to be the same, weigh the same, and may perhaps have come from the very same refinery.

The fuel companies tried chemical analysis in an endeavour to achieve parity from one batch of petrol to the next, but in spite of intensive lab programmes they were not able to identify the batches that were prone to promote engine knock. Because of this problem special fuel research engines, with a variable compression feature, were constructed to evaluate and grade fuels. Such a standard heavy-duty single cylinder test engine would be warmed up to a standard test temperature, run at standard rpm and load, and then have the compression ratio increased until the fuel being tested just produced engine knock. Its anti knock rating would then be specified as its Highest Usable Compression Ratio (HUCR).

However, even with every fuel lab supplied with the same type of test engine and using the same standard test procedure, it was discovered that the same fuel could test out with differing HUCR numbers in different laboratories. Obviously some

The standard fuel test engine is a heavy-duty single cylinder unit. Insets show the rod and piston, which take a regular pounding without failure, and the knock meter.



unvarying standard was needed by which to calibrate the lab test engine. Two pure substances were chosen as reference fuels. The high reference fuel was what we commonly call iso-octane (2-2-4 trimethylpentane), while the low reference fuel was normal heptane (n-heptane).

Now it was decided that a fuel under test would be run in the variable compression test engine and its HUCR determined. Then a series of runs would be [illegible] supplied for a variety of applications.

RESEARCH AND MOTOR TESTS

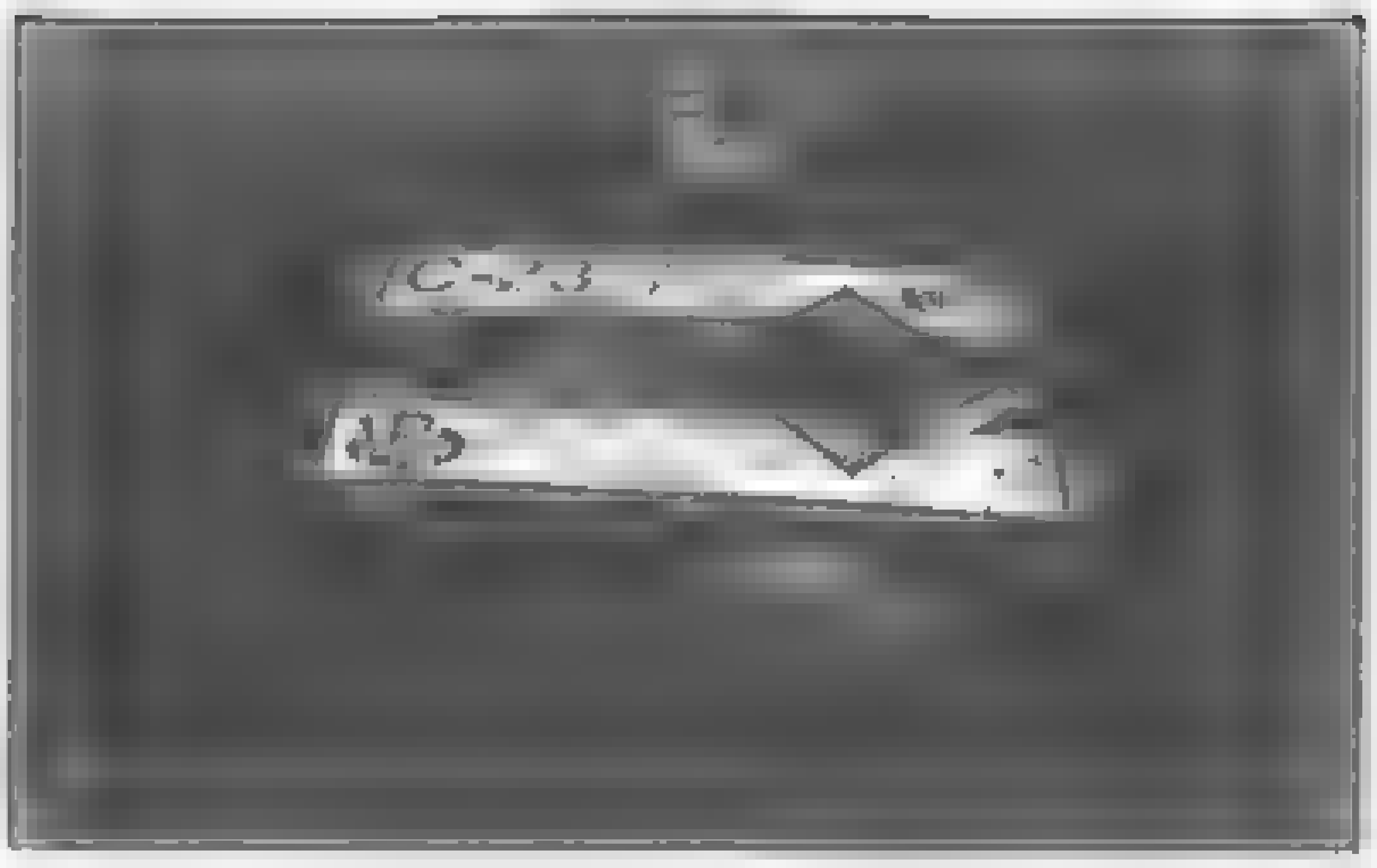
Since that time a number of test procedures have come into use to simulate a variety of [illegible] test methods. Both measuring techniques use the same single cylinder, variable [illegible] greater engine speed and a higher inlet mixture temperature. Hence the Motor method is a more severe test, and generally yields octane numbers 6 to 12 less than the Research numbers as shown in Table 6.2. This distinction is important as it informs us [illegible] temperatures for long periods at full throttle. Fuel technicians often state that the RON gives a better indication of a fuel's part throttle knock resistance, while the MON indicates its full throttle knock resistance.

Table 6.1 Comparison of Motor and Research test procedures

	Motor Octane Test	Research Octane Test
Inlet air temperature	148.9°C	65.6°C
Engine jacket temperature	100°C	100°C
Engine rpm	900	600

Table 6.2 Octane test comparison

Research octane number	Motor octane number	Pump octane number
92	85	88.5
96	88	92
98	90	94
100	91.5	95.8
105	95	100
110	100	105
113	103	108
115	105	110



VP racing C-23 has a very high Motor Octane Number (MON) of 119. It is l d (7.2gm/gal) and has a specific gravity (SG) of 0.71

The spread between the RON and MON is known as the fuel's sensitivity an is quite important to understand what this distinction is exactly. Because intake temperature affects different fuel compounds in various ways it is possible for a refiner to come up with a fuel which has a high RON (or high Pump Octane Number) of sa but by the Motor test that same fuel would rate as, say 86 octane. Hence it would perform as an 86 octane fuel in a typical performance engine. However, company could use a different blend of fuel compounds that was in storage or the type of crude oil being 'cracked', and produce RON of 98, but with a MON of 91. This is one reason for the frequent compl among performance enthusiasts of 'bad fuel'. The RON may really be up where the company say it is, but because of the sensitivity of the blending compounds could be performing like low octane stuff when high lead levels were usin this sensitivity problem seldom surfa tends to 'cushion' fuel sensitiv However, with today's unleaded and le d replacement fuels, fue sitivity wi continue to cause us problems for as lor RON figures are used as the industry standard to rate commercially availabl

R + M/2 AND SUPERCHARGE NUMBERS

In the USA service station pumps carry a Pump Octane Number (PON) rather than th Research number seen on pumps in other parts of the world. Th xpressed as R + M/2 is the average of the RON plus MON divided by tw

The Supercharge test is applied to aircraft fuels which exceed 100 umbers, as the other tests obvio ss at anythin ane. The Supercharge Octane Numbers (SON) are really performance n btained by linearly extending the scale beyond 100. In this test the reference fuel i so-octane plus lead (1gm of lead added to 4 litres will increase the octane about points). Two tests are involved, the F3 and F4, which explains wl fuel rating such as 100/130. The first number refers to the F3 test which simula percharged engine running on a chemically correct air/fuel mixt

cruising. The 14 number gives an indication of the fuel's performance rating with enriched mixture and increased boost, as would be supplied during take-off.

FUEL CHEMISTRY AND OTHER TEST STANDARDS

The anti-knock properties of hydrocarbon fuels are related to their molecular [illegible] from the combustion flame itself. Therefore when tightly bonded molecules are [illegible] rate of pressure rise and combustion speed to an orderly controlled process.

Another factor in all of this is the rate at which fuel vaporises, and at what temperature. If the fuel recombined in the inlet tract or cylinder to the liquid state it wouldn't burn. On the other hand if it boiled easily into vapour, controlli[illegible] fuel with the right mix of compounds to provide the necessary knock resistance (octane level) and which readily vaporises at specific temperatures.

In the past a lot was made of a fuel's RVP (Reid Vapour Pressure), but relative to fuel performance the numbers are pretty meaningless. Governments regulate the RVP to ensure relatively safe fuel handling (if it boiled too easily fuel tanks and drums would bulge and eventually explode), reduced emissions during fuel transfer [illegible] in summer, and between 7.98 and 13.05 in winter. Formula 1 fuel regulations demand RVP [illegible] 7psi, and most race fuels fall into the same range.

Leaded 100 racing fuel is often straight Avgas 100/130. This is the high lead green variety, but as the drum states that it is not for aviation use. Other hydrocarbons – most likely benzol and/or toluol, which would have lowered the RVP below the aviation standard – have obviously been blended in.

What gives us much more insight is the distillation curve as it shows us how rapidly and at what temperatures the fuel turns to a readily combustible vapour. The volume which evaporates up to 70°C must be adequate to aid cold starting, but if the proportion is too large fuel will boil in the fuel lines. The amount which vaporises at 100°C establishes the engine's warm-up running qualities, and also its acceleration and throttle response at normal engine temperature. The vapourised volume at 140°C and 180°C should be sufficiently high to minimise dilution of the lube oil, particularly before the engine reaches normal operating temperature.

The European Standard for summer blend states that at 70°C a minimum volume of 15% must evaporate. By 100°C at least 40% has to be boiled off. At 180°C 85% of the fuel must have vaporised, and by 215°C all must have boiled to vapour. By comparison Formula 1 fuel, which is supposed to mirror what you can purchase at European service stations (I kid you not!) has to have lost a minimum of 20% volume to vapourat 70°C, while at 100°C 46% has to be boiled off and at 140°C the minimum which has boiled off [illegible] 90% and 95% of the fuel is vaporised, and the final boiling point (FBP) – the temperature at which all liquid has been converted to gaseous form (Table 6.3).

Table 6.3 European Standard race fuel characteristics

		Elf Avgas 2T	Elf WRF	Elf LMS	Elf Moto 4T-2M
Octane RON		118-120	101.6	101.2	99
MON		102	89.4	89.4	86
PON		110-111	95.5	95.3	92.7
Specific gravity		0.784	0.763	0.765	0.765
RVP		3.10	6.96	6.96	7.25
Distillation	30%	95°C	76°C	82°C	70°C
	50%	100°C	99°C	101°C	82°C
	90%	106°C	128°C	134°C	119°C
	95%	108°C	151°C	158°C	
	FBP	120°C	169°C	173°C	140°C
% vol @ 70°C			24	20	
% vol @ 100°C			52	50	
Oxygen content		2.4%	2.6%	2.5%	2.6%
Lead content gm/gal		2.27	0	0	0

Elf Avgas 2T 96.7 – complies with regulations permitting Avgas 100LL fuel
Elf WRF – control fuel for World Rally Championship
Elf LMS – control fuel for Le Mans
Elf Moto 4T-2M – complies with Superbike regulations

In America fuel regulations are different. As such their fuel test sheets will not include details of the volume of fuel vaporised at 70°C and 100°C. They record at what temperature fuel first evaporates, called the IBP (initial boiling point). Then it is recorded at what temperatures 10%, 50% and 90% of the fuel turns to vapour. Finally they record the end point, the temperature at which all the fuel changes to vapour.

rating

Table 6.4 American Standard race fuel characteristics.

	Trick 108	Trick Turbo	VP MR#9	VP MS 103	VP C-23
Octane RON	113	119	98.1	107	
MON	103	109	87	99	119
PON	108	114	92.5	103	
Specific gravity	0.73	0.73	0.718	0.743	0.71
RVP		6.1	11.22	2.86	4.54
Distillation IBP	36°C	34°C			
10%	58°C	68°C	44.4°C	87.8°C	77.9°C
50%	94.5°C	96°C	57.7°C	99.1°C	97.6°C
90%	117°C	113°C	115.6°C	102.8°C	99.7°C
end point	146°C	144°C	164.8°C	107.6°C	110.1°C
Oxygen content			2.7%	MTBE	
Leaded	Yes	Yes	Yes	No	Yes

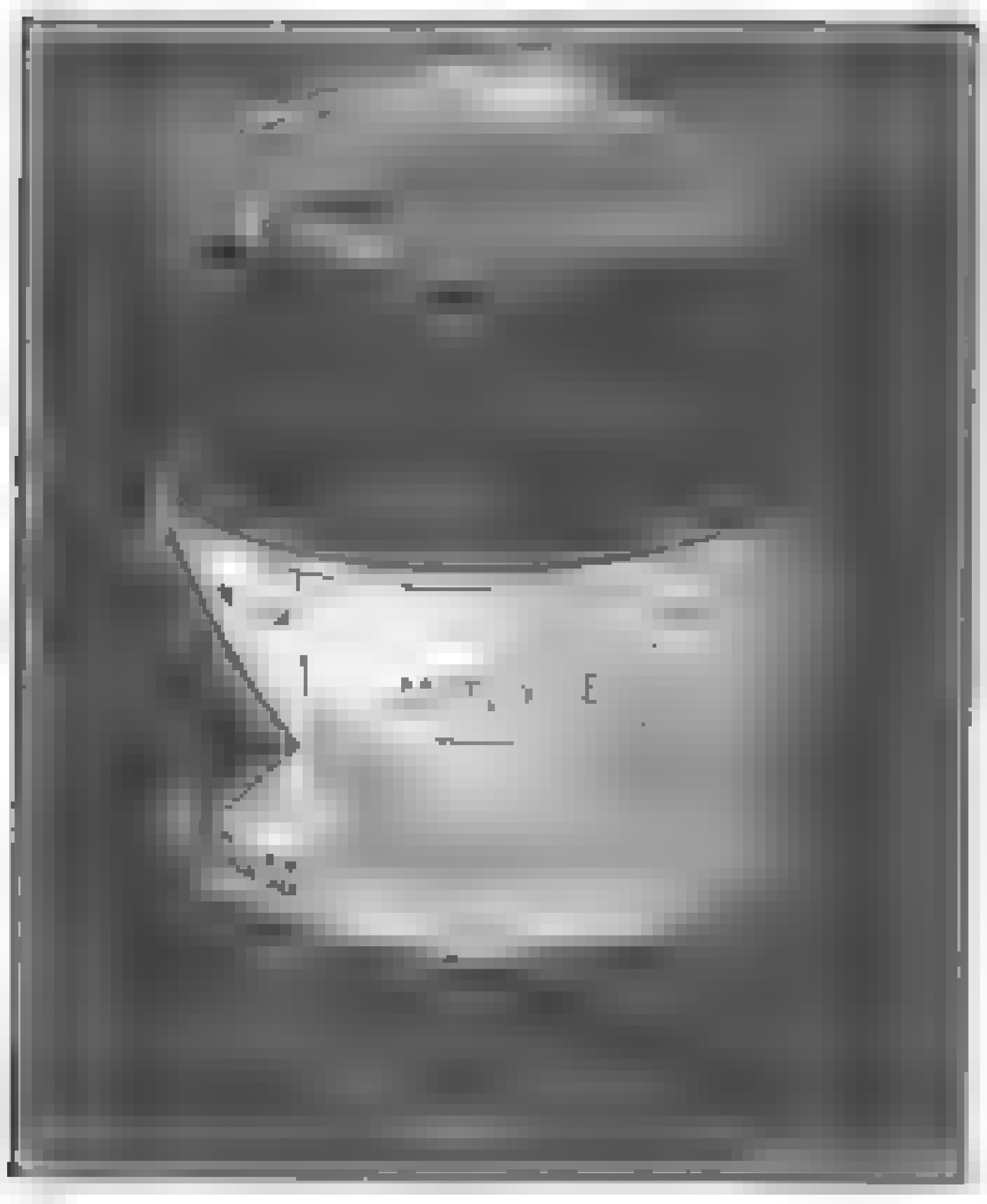
companies actually change the formulation each month. In aircraft the throttle is

running with infrequent start-ups, cylinder lubricant wash and oil dilution isn't an issue. In this regard race fuels are similar, but unlike aircraft race cars must have

knock fractions and more easily ignited components

LOW MOLECULAR STRUCTURE AFFECTS KNOCK RESISTANCE

The paraffins, such as normal heptane and kerosene, are long chains of carbon and hydrogen held together by weak molecular bonds, which are easily broken by heat



Methyl Benzine, Shell's name for toluol, is [illegible] *high knock resistance. During the Formula 1 turbo era the us*[illegible] *blend was 86% toluo*[illegible] *14% n-hepta*[illegible] *being added to improve fuel atomis*[illegible] *combustion spee*[illegible]

Iso-octane is a member of the iso-paraffin family. These have branched chain structures that form stronger bonds to better resist detonation. The cycloparaffins or napthenes also have good anti-detonation properties with their hydrogen and carbon atoms well bonded in a ring shaped molecule. The aromatic fuels, such as toluol, also have a ring-shaped structure with very strong bonds, hence their excellent anti-knock characteristics.

As has already been pointed out, the chemical composition determines just how rapidly and completely the fuel will burn, and whether it will be resistant to detonation at high cylinder pressures and temperatures. The fuels required for ease of starting and throttle response have weak molecular bonds, so break up and burn spontaneously (ie without being ignited by the combustion flame initiated by the firing of the spark plug) at lower temperatures and pressures than fuels with strongly bonded structures. Some fuel additives, such as the aromatics, make good anti-detonants because they burn slowly and don't oxidise or burn completely until combustion chamber temperature and pressure is very high. Such components thus inhibit, or slow down, combustion. Hence a high octane fuel will only increase power in an engine that actually needs a fuel which is chemically stable at high temperature and pressure. Obviously if [illegible] engine does not have a compression ratio, boost pressure and spark advance [illegible] enough to produce high combustion pressure and temperature, then the [illegible] fuel may not burn completely during the early phase of the power stroke, resulting in reduced throttle response and perhaps [illegible] power loss.

This is one reason for my fondness for the dual-fuel approach in [illegible] performance road cars. With an ordinary, relatively inexpensive street fuel in the main tank you get easy starting, good throttle response and minimal cylinder wash and oil dilution. Then with either toluol, a toluol/Avgas mix, or straight Avgas in the secondary tank, connected to [illegible] second injection system, you achieve excel[illegible] detonation resistance with inexpensive, easily obtainable fuels. Plus, because these fuels with tightly bonded structures are only being released at higher engine [illegible] levels, combustion is rapid and complete, ensuring good power output combined [illegible] high level of knock resistance.

AVGAS-BASED RACE FUELS

In the past a lot of race fuel was simply some grade of Avgas relabelled. Thus 115/145

aromatics, usually toluol

Avgas 112/160 used to be available in some parts of the USA and is now marketed

race car to go another 5–6% distance, an important factor in endurance events.

SOPHISTICATED RACE FUELS

the top of the power curve and even more in the mid-range, which also serves to

Motorsport 103 is a street legal oxygenated RFG unleaded fuel with a high octane rating (103 by R+M/2 method and 99 MON) and SG of 0.728

broaden the power band, are seen on the dyno. MR#3 from VP is representative of this class of fuel. It is heavily leaded (7.2gm/gal), contains 2½% oxygen, has a seemingly low MON around 90, but its actual resistance to detonation under race conditions makes it seem 4–6 numbers higher, and costs $US 7.00 per litre!

While this was going on more sanctioning bodies began to ban leaded fuels in the more highly promoted and publicised classes (NASCAR being the exception). The transition has not been too smooth for some racers though. What has caused some confusion is that 'Unleaded Racing 100', which in Europe is rated by the Research method, unlike the old leaded 'Racing 100' which gets its 100 rating with the tougher Motor test. Obviously a switch to the unleaded 100 without suitable engine modifications and retuning would cause the engine to be destroyed by detonation.

Adding to the problem was the fact that many high octane unleaded road and race fuels contain the quite effective octane booster MTBE (methyl tertiary butyl ether) in concentrations usually below 10%, but sometimes up to a maximum of about 15%. MTBE is also an oxygenate, meaning it carries some oxygen, so as well as giving an octane boost it also provides a tiny amount of additional oxygen for the combustion process. So it is important to note that if the race fuel specifications indicate an oxygen content that this is taken into account as it affects fuel/air ratios. MTBE is more dense than petrol, so the SG goes up. Previously a high SG would alert a tuner to lean off the quantity of fuel going into the engine. However because MTBE is an oxygenate the SG is a somewhat misleading measure as to how much the mixture can be leaned off. Some tuners did not understand this and lost engines. In America, VP Motorsport 103 is very popular (99 MON, 107 RON), it contains MTBE but VP do not state the oxygen content (they do on most other fuel). The power king in this class is Power-Mist RFG (reformulated gasoline). It contains 6% oxygen and tests at 105 MON and 111 RON.

By far the most money and resources are devoted to Formula 1 fuel development. F1 must use unleaded with maximum RON of 102 and maximum SG of 0.775. Maximum oxygen content is now 2.7% and benzene allowable is right down to 1%. Even after the turbo era aromatics in F1 fuels were often over 50%, but now have been pegged at a maximum of 35%. Major components in descending order are: toluol, xylene, isopentane, iso-octane, hexene-1, n butane, 2-methylpentane.

OCTANE BOOSTERS

As you can see toluol and xylene are the main anti-knock agents in F1 fuel. Unfortunately straight xylene is difficult to obtain (paint thinners may contain a considerable portion of xylene and/or toluol but as it also contains other compounds its use is fraught with danger) and while such is not the case with toluol there is a question mark over using it as an octane boosting agent in high octane unleaded pump fuel. In the past a reliable method of increasing the octane rating of petrol was to add up to 33% toluol (bigger percentages were not recommended due to cold weather starting difficulties). This percentage would raise the octane rating of leaded fuel (0.2 gm/ltr) about 6 numbers Research and 2.5 numbers Motor respectively. However, as lead levels have decreased, government regulations permitting, petrol refiners have substituted high-octane hydrocarbons to bring the octane back up, so a reliable octane increase cannot be guaranteed by their addition. A guide as to whether fuels such as toluol, triptane, benzol, xylene etc have been added by the fuel company is to check

Concentrated octane boosters vary cons... degree of caution

the specific gravity. If the SG is high and there is no government regulation limit these products to 5% maximum (they are reported to be cancer causing), the petrol probably contains up to 20% of these products either singly or in combination.

However if the government limits their use to a maximum of 5% then with the exception of benzol and triptane all will provide a good octane boost of about 1 MON for every 10% added up to a maximum of around 25%. Additionally, in distance racing, by virtue of their high SGs they significantly reduce fuel consumption ... thus the number of pit stops required.

The situation is similar with another very effective octane booster, MMT (methyl cyclopentandienyl maganese tricarbonyl), which is the base product in the most impressive concentrated octane boosters (Australian made NF Racing Formula and Nulon Pro Strength). In low octane unleaded (91 RON) both will provide a consistent 5 RON boost. However when added to 95 RON premium unleaded the results waver between a 2 and 3.5 RON rise, and with super 98 RON unleaded it varies between a 1.5 and 3 RON increase.

... e increase claims. If any of the claims ... e been obtained when added to the 83–86 RON stuff which is ... for motor vehicles in some ... ts of the earth. Also be careful of ... in product which you have previously found to be effective. The ... 'formulated', new packaging etc. should get the alarm bells ringing. It probably means the product has had to be changed due to government regulations or steep increases in the price of key ingredients, and that it no longer is as effective as previously. 104+ and Super 104+ are a typical example. The old stuff was about best of the 'bottle boosters'. Then in 1999 it was changed (identified by the ... back) and slumped to being a very ordinary performer giving only about half the boost (1 RON for Super 104+ with 95 RON unleaded ... its previous form ...

When using these concentrated additives there are two points you must keep in mind. The first is that American concentrates are usually marketed in US quarts which equal 946cc as opposed to the Imperial quart of 1136.5cc. The US gallon is also different (see Appendix), so if you live outside the USA you will have to carefully work out the blending ratio. The second point is with regard to mixing technique. Don't just pour a can of octane booster into your fuel tank and expect it to blend uniformly. What is preferable is to mix the concentrate with about 2–3 gallons of petrol, give the mixing drum a good shake, and then add this mixture to the untreated fuel.

TESTING, PURCHASING AND STORAGE OF PUMP FUEL

Really any time you blend a commercial pump petrol you are at the mercy of the fuel companies, so what worked with one batch of fuel may not work with another. Apart from certified racing fuels the only other fuel that is a known quantity and highly regulated is Avgas. Consequently my advice is don't muck around with fuel blending, it could cost you an engine. If you are in a competition class restricted to 98 RON unleaded test different brands to determine what brand gives the best overall power and throttle response. Do this preferably in summer when fuel turnover is high and when summer blend is being supplied. When you find the 'power king' buy up what you require to do you for the race season. Be sure to store fuel in sealed drums away from sunlight, you don't want vital components evaporating or being destroyed by light (both can happen and high octane unleaded is more prone to this damage). Also remember fuel companies swap fuel between themselves and frequently use a series of common pipe networks so if your dyno testing of fuel is going to be of any value you really have to be quick to buy fuel from the very same service station where you sourced your winning test sample before new, possibly different supplies, are dumped in the storage tanks.

WHY METHANOL RAISES HORSEPOWER

By now it should be obvious that a fuel's octane rating is not a true measure of its power potential. Its octane rating, its knock resistance, is important but it is not the whole story. When we examine an alcohol fuel such as methanol this becomes even more evident. When we look at Table 6.5 we see that methanol has only a very modest Motor octane of about 89–91 and the high Research numbers are only possible with extremely rich mixtures. The problem then is, why does it permit a higher compression ratio (CR) than petrol and provide up to a 20% power increase?

Methanol has a very high latent heat of vaporisation,

converted from liquid into vapour. Petrol has a latent of evaporation of 135Btu/lb, while for methanol it is 472Btu/lb (One British Thermal Unit (Btu) is the amount of energy required to raise the temperature of 1 pound of water 1°F). This heat required for proper atomisation is drawn from the inlet tract, the piston crown, the combustion chamber and the inlet and exhaust valves. An internally cooler engine does not put as much heat into the inlet charge, with the result being that mixture density in the cylinders goes up, and hp rises. Also less initial heat means a slower burn, so the engine may accept more spark lead or higher CR, resulting in more hp or improved throttle response.

Table 6.5 Fuel characteristics

Fuel	Specific gravity	RON	MON	Fuel/air ratio (lb/lb)	Heat energy (Btu/lb)	Latent heat of evap	Weight (lb/g
Acetone	0.79			1:10.5	12,500	225	8
Avgas 100/130							
'green'	0.69	105-110	100-102	1:12.9			7
'blue'	0.71	105-110	100-102	1:12.7			7
Benzol	0.88	105-110	95-100	1:11.5	17,300	169	8.7
Ethanol	0.79	108-115	90-92	1:6.5	12,500	410	8
Ether (diethyl)	0.71				15,000	153	7
Methanol	0.79	105-115	89-91	1:4.5	9,800	472	8
Nitromethane	1.13			1.2	5,000	258	11.3
Nitropropane	1.05				6,700		10.5
Petrol							
premium unleaded	0.74	*96	*85-86	1:12	19,000	135	7.4
premium leaded	0.73	96	86	1:12.5	19,000	135	7.3
racing leaded (USA)	0.73	112-114	102-104	1:12.7			7.3
racing unleaded (USA)	0.75	104-106	94-96	1:13.2			7.5
racing unleaded 100	0.75	100	90-92	1:13.0			7.5
Propylene oxide	0.83				14,000	220	8.2
Toluol (methyl benzine)	0.87	120-124	110-112	1:9.8			8.7
Triptane	0.69	110-112	100-102				6.9
Xylene	0.86	117-118	115-116				8.5

* Premium unleaded in some lands has a higher rating (often 98 RON, 87 MON)

That's part of the story; the other part relates to the amount of energy available in the fuel [illegible] somewhere between 12.3:1 and 13.1:1, with about 12.7:1 not far from the mark. With methanol the air/fuel ratio increases to somewhere between 4.5:1 and 5.5:1, with about 5.1:1 being pretty close. One pound of petrol has the energy potential of about 19,000 Btu. In [illegible] with each pound of air, we are actually producing more heat energy by burning methanol.

To work out how much more heat energy is produced we have to divide 12.7 by 5.1, which is 2.49. Next we multiply 9,800 by 2.49, which gives 24,402. This tells us that in theory methanol will produce 28% more heat energy than petrol

$$\left(\frac{24{,}402 \times 100}{19{,}000}\right) - 100 = 28.4\%$$

On average this equates to a power increase around half that, or 14%. Some engines will give close to 20% more hp, particularly if they have been prone to overheat burning petrol, while for some others the gain will be closer to 10%.

Are alcohol fuels only for high-compression race engines?

FUEL FLOW INCREASES REQUIRED WITH METHANOL

Right away you can see that an engine running straight alky will burn more than twice as much fuel as one burning petrol. Therefore you must be careful to ensure that the needle valve, fuel pump and fuel lines will flow the required amount of fuel.

This can present some problems as many carburettors will not flow the required amount of fuel through the needle and seat, and others do not have main jets large enough. For this reason fuel injection is preferred for alky and nitro burners.

The SU carburettor can be fitted with two or even three fuel bowls if necessary to get proper flow. Holley and Weber have alcohol size needle valves available. If you have a carburettor that you cannot get larger needle valves for, you will have to enlarge the discharge area of the standard needle and seat, but check the fuel flow into a container before you run the engine. If it is not flowing enough fuel you will melt the pistons.

Some carburettor jets are classified with regard to the fuel flow, the number stamped on the jet representing the ccs of fuel passing through the jet in a given time. For example, the hexagon-head Mikuni carburettor jets follow this pattern. If you are

Shell 'A' racing fuel is 96% methanol and 3% acetone

changing from petrol to alky then you will have to start testing with jets 2.2 times large, eg change a 200 main jet to a 440.

excellent set of drills for this purpose, but they are expensive.

PROBLEMS AND DANGERS OF METHANOL

Caution must also be exercised in other areas of the fuel system. Petrol tends to

dead rider due to an aluminium or magnesium slide stuck wide open.

In colder climates, starting difficulties may be encountered when straight alcohol

overcome this problem, usually 5% acetone or a maximum of 3% ether. Ether is

being cranked over, if it backfired you could be badly burned.

Alcohol fuels absorb water so either a stainless steel or triple marine grade anodised aluminium fuel rail and methanol injectors are essential

before fitting the warm-up spark plugs, spray about 3–5cc of petrol down each spark plug hole

Many starting problems can be traced to an inadequate ignition or electrical system. Obviously the system must have enough grunt to spin the engine rapidly to fire. Also the ignition system must be up to firing wet plugs or fuel loaded spark [illegible]. Methanol burns much more slowly than petrol, so it is necessary to advance the ignition accordingly. As a guide you can reckon on the engine requiring 15–25% more advance than for petrol, right through the rev range

NITROMETHANE USE AND BLENDING

Nitromethane is different again, and if you look at the fuel characteristics table (Table 6.5) you will see that it is not even assigned an octane number (because there are too many [illegible]

methanol, for every 20% nitro added it increases hp by 10%. This is possible due to nitro's very special chemical composition, and also due to the fact that extremely rich mixtures can be burned satisfactorily. In itself, nitromethane is a very poor fuel, but because it contains approximately 53% oxygen by weight it permits the induction of large quantities of fuel into the engine for conversion to heat energy. Also because it burns slowly it continues to apply force to the pistons almost to the bottom of the power stroke.

In the past nitromethane was burned in the most powerful drag machines in ratios of 80–98% nitro and 2–20% methanol. Today Top Fuel rails are limited by the rules to 55% nitro in an attempt to restrict engine power and trap speed. On the speedway scene and hill climbs nitro is used, often illegally, in smaller percentages (usually 10–20%) as a power booster. Methanol is the base fuel, and acetone plus other 'doping' agents may also be added (plus about 0.5% castor oil [illegible] an odour masking agent

To deter detonation or other engine damage, it is always necessary to reduce the compression ratio when nitro is used. At all times the air fuel ratio must be very rich. With an 80–90% blend of nitro it may be as rich as two parts fuel to one part air, or a

can as two parts air to one part fuel. Using 20% nitro this would change to three or four parts air to one part fuel (Table 6.6).

Table 6.6 Fuel flow increase for nitromethane

% nitro in methanol (by volume)	Flow increase over straight methanol
10	25%
20	50%
30	75%
40	100%
50	125%
60	150%
70	175%
80	200–230%
90	225–260%
98	250–290%

When blending nitro with other fuels the safest method to avoid error is to mix according to volume. Thus for a 20% nitro blend you would add one litre of nitro to four litres of methanol. Some people also blend using a hydrometer calibrated for this purpose. The problem with this method is that the hydrometer will only give a true nitro content percentage reading when the fuel temperature is 20°C/68°F. From Table 6.7 you can see that with a fuel temperature of 38°C/100°F, a fuel with a true nitro content of 80% will read 73% nitro on the hydrometer. Consequently you should always measure the temperature of the fuels before attempting any mixing.

Many don't seem to understand that a nitro hydrometer is calibrated to give a true reading only when straight methanol and nitro are being blended. When a [illegible] of no value at all.

Table 6.7 Nitro hydrometer temperature correction chart

% nitro	Hydrometer reading at fuel temperature °C/°F										
	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
100	106	104	102	100	98	97	95	94	92	90	87
90	97	94	92	90	88	87	86	85	83	80	78
[illegible]	86	83	82	80	78	77	76	75	73	70	68
[illegible]	75	73	71	70	69	68	66	65	63	61	59
[illegible]	66	63	61	60	59	58	57	56	54	52	50
[illegible]				50	49	48	47	46	44	42	40
[illegible]				40	38	37	36	35	33	31	30
[illegible]	35	33	31	30	28	27	26	25	23	22	20
[illegible]	27	25	[illegible]	20	19	18	17	16	15	13	11
[illegible]				10	9	9	8	7	5	3	1

Igniting high percentage nitro fuel is always a problem. Top Fuel dragsters run the equivalent of a mini welder delivering 1.2 amps to each spark plug. As higher percentages of nitro are burned the combustion rate is reduced (unless a fuel ignition accelerator such as propylene oxide is added). Therefore the spark advance must be increased, with Top Fuel engines typically running a system locked on about 50° advance.

NITROMETHANE DANGERS

There is considerable risk to yourself and others associated with the use of nitro. After combustion relatively large amounts of vaporised nitric acid are exhausted. The higher the nitro dose, the more acid vapour is released. When inhaled, nitric acid vapours cause a muscle reaction, making it impossible to breathe. Therefore the use of the correct gas mask is essential if the driver is in a position to inhale the exhaust gases. Certainly mechanics working around the car and those in the starting area will require masks.

Some people have the idea that nitromethane is explosive. It isn't, but like any fuel it can be made shock sensitive. The following are the main causes of nitro becoming dangerous.

- The addition of hydrazine in fuel blending. Hydrazine is banned in many countries because of the danger.
- The use of caustic soda or any other alkaline for cleaning fuel tanks or lines.
- The use of 'unpickled' anodised aluminium fuel tanks. After anodising the tank must be allowed to stand for a few days filled with a solution of 90% water and 10% vinegar. This serves to remove any deposits remaining in the tank after anodising.
- The use of excessive fuel pump pressure. Nitro is liable to become unstable when confined and subjected to shock. Even though some Top Fuel dragsters routinely run 500psi fuel pressure anything over 100psi should be considered dangerous.

When using more than 20% nitro, there is always the danger of a sump fire or explosion due to the large amount of fuel that finds its way past the rings and into the sump. The first signs of such a fire are yellowish flames appearing at any of the breathers. Therefore it is important to keep an eye on the engine for at least 2–3 minutes after shut-down.

OTHER FUEL COMPOUNDS

All the base fuels – petrol, methanol and nitromethane – may have other compounds added for a variety of reasons.

Propylene oxide (epoxy propane) is used in high percentage nitro fuel to increase the combustion flame speed. The amount used should never exceed 15% of the amount of nitro in the fuel as the combustion rate could rapidly increase to the point of severe detonation and engine destruction. For example, if you are using 80% nitro you could use up to 12% propylene oxide.

Propylene oxide can also be used with Avgas or heavily leaded race fuels, and also methanol up to a maximum of about 5% to give a modest 2-3% power gain. Of much greater benefit though is the greatly improved throttle response which makes the [illegible] However, caution is in order to get the mixture a little rich and the timing backed off, otherwise expensive engine damage will result.

Once blended with other fuels, propylene oxide is relatively stable, but it can [illegible] copper, copper alloys, or rust particles. Therefore it must be stored in aluminium or plastic containers.

Nitropropane is listed in Table 6.5, but I would recommend its use, with extreme [illegible] and will cause a severe lean-out unless fuelling is enriched. As a starting point, increase the fuelling by three times the percentage of nitropropane added, (ie for 10% nitropropane increase fuelling by 30%) and take out some spark lead too as combustion speed with this stuff can quickly get out of hand.

Like propylene oxide, acetone and picric acid, nitropropane is a member of the nitroparaffin family. It can be used with heavily leaded race fuel or Avgas up to a maximum concentration of about 12%, to produce a power gain in the order of 5-6%. It can also be blended with methanol in like percentages to provide a similar power boost as that when added to Avgas. However, if more than about 3% acetone (propanone) is present in the methanol then concentrations over 5-6% should not be used. Be aware that many fuel companies do not supply pure methanol. Often they blend in 2-3% acetone to improve methanol's ability to be mixed with other fuel compounds. Actually up to 10% acetone will improve the combustion rate of methanol, improve cold weather starting, and in like vein to water reduce the tendency for lean mixtures to detonate.

ASSESSING OCTANE REQUIREMENTS

Some have stated that an engine requires an octane increase of 5.0 for every 1.0 increase in the compression ratio. At best this is a gross oversimplification of a very complex matter, at worst such a 'rule' could easily lead to serious engine damage. I would prefer to state the matter in quite different and non-specific terms.

The first 'truism' is that anytime we significantly increase the intake air temperature, and place the engine under load we need increased fuel octane to ward off detonation. Increases in ambient temperature obviously produce similar increases in intake air temperature. Unscavenged exhaust gas does the same thing, but actually inside the cylinders. Increased water and oil temperatures also add heat to the inlet charge. The latter can be the product of higher ambient temperature, but it can also be as a result of a very dense, cool inlet charge producing a massive increase in heat energy as a result of all the fuel added to that dense charge to provide the correct air fuel ratio.

Now I have to make what may appear to be a reversal of my first truism, that is anytime we significantly increase charge density, and place the engine under load, we need additional fuel octane to fend off detonation. Thus cool dense atmospheric conditions could produce engine knock. In similar vein adding an efficient 'cold air' or

Here the combustion chamber volume is about to be measured so that the compression ratio can be calculated. While the compression ratio is important in assessing octane requirements, other factors such as engine temperature, cam lobe characteristics and cam advance, combustion chamber sh[illegible] [illegible]ust efficiency, vehicle weight and [illegible]ng, engine rpm, etc. all have some influence.

ram air system could produce engine knock. As indicated in the sidebar a big drop in humidity could lead to the engine detonating. And, yes, a more efficient exhaust could cause knock. Less unscavenged exhaust gas increases ch[illegible]e density, but less exhaust gas means less corruption of the combustion flame through the mixture. More flame speed can lead to an excessively rapid cylinder pressure rise, out of cont[illegible] combustion – detonation.

As I stated this is quite complex. It is all very much a delicate balance. A s[illegible] increase in charge temperatures or charge density isn't a problem, but a big inc[illegible] [illegible]ther will call for more octane. Likewise a small increase in both ch[illegible] charge temperature will likely require mo[illegible]

PERFORMANCE INCREASES WITH RACE FUELS

Finally let's consider in real terms how m[illegible] [illegible]ed race fuel can be worth [illegible] when an engine can't be re-tuned to ta[illegible] much advantage of the increased knock resistance and superior combustion characteristics. The test mule was a stock Hon[illegible] VTR-1000 SP-1 from a race series restricted to strictly standard 1,000cc V twin bikes. The engines and electronics were sealed to prevent any tampering, control tyres w[illegible] supplied and only brake pads, dampers and springs were free. Consequently with fre[illegible] starting grids on some relatively sho[illegible] circuits regular podium finish[illegible] be hard to achieve. The series controllers encouraged the use of 98 RON pr[illegible] [illegible]ed, and to prevent cheatin[illegible] [illegible]pecial 'home brews' any runners electing t[illegible]

Humidity – a natural octane booster

[illegible]

Percentage humidity

[illegible]

to source their fuel from the circuit pumps were required to use only race fuels not containing any alcohol or nitro. The race fuel had to be decanted directly from a sealed drum into the bike's fuel tank, and fuel samples taken at any time from the bike had to match a fuel sample that the organisers had drawn from a sealed drum prior to the race season commencing. A further disincentive was that all fuel testing costs were charged to the competitor.

Given the series promoter's tight fuel policy and the bike's stock tune requirement many competitors simply resigned themselves to the vagaries of circuit pump fuel. The reasoning being 'how much can race fuel be worth in a stock road engine that cannot even have an ignition advance adjustment?' A few though saw the fuel rule as a big hole in the series regulations and feared that they could lose out if they didn't at least investigate if there were any gains to be made. My feeling was that as these engines were operating at up to 11,000rpm and cylinder diameter was close on 100mm there was a real possibility of finding a fuel with superior characteristics that

enabled it to firstly, vaporise and blend better with the inlet air charge, and secondly, to burn more completely. Thus providing a power boost and better throttle response.

With a bore of 100mm and a relatively short stroke of 63.6mm the Honda could have been at a disadvantage in the mid-range compared to other manufacturers. Its 54mm throttle body was larger than the rest, possibly giving rise to sluggish throttle response. At 10.8:1 its compression ratio was 0.6–1.0 lower too, so there was a real need to find gains in engine performance.

Table 6.8 Race fuel test

rpm	Test 1 hp	Test 2 hp	Test 3 hp	Test 4 hp	Test 5 hp	Test 6 hp
4,500	58.6	55.4	59.7	58.3	63.2	58.9
5,000	61.3	59.5	62.1	61.4	65.8	61.5
5,500	62.9	61.2	64.0	62.8	66.7	63.8
6,000	76.8	74.5	76.7	76.8	78.2	76.8
6,500	85.4	83.9	85.2	85.6	88.1	85.5
7,000	93.0	92.6	93.3	92.9	96.3	93.4
7,500	102.4	102.5	102.6	102.3	106.0	102.7
8,000	110.5	110.6	110.3	110.8	114.9	112.1
8,500	115.3	114.9	115.9	115.6	119.7	116.8
9,000	118.1	118.2	118.9	118.4	122.6	121.8
9,500	116.6	116.8	117.0	116.9	120.9	119.5
10,000	113.9	113.9	114.4	114.2	118.9	116.3
10,250	107.3	107.4	107.8	107.6	112.1	111.6

Test 1 – Pump 98 RON summer blend unleaded
Test 2 – Racing 100 leaded (Avgas 100/130LL)
Test 3 – Racing unleaded 100 MON
Test 4 – Racing leaded 108 MON
Test 5 – Racing leaded 90 MON oxygenated
Test 6 – Racing unleaded 105 MON heavily oxygenated

Table 6.8 shows the results from some of the fuels tested. As expected Avgas 100/130 was the poorest performer in hp terms, and throttle response on the dyno felt pretty 'doughy'. What was a real surprise was that the majority of expensive specialty race fuels, sourced from Japan, USA and Europe, were only marginally better, both in outright power and throttle response. Obviously the outcome would have been quite different if ignition advance and fuel curve tweaks had been permitted to take advantage of the higher octane. However one fuel did stand out (Test fuel No. 5) and a couple of others also worked well (Test fuel No 3 at the bottom of the power curve and Test fuel No. 6 at the top of the power curve).

When testing was transferred to the race circuit Test fuel No 5 proved clearly superior. No 3 was only better than 98 RON pump fuel on a fast track where full throttle was held for a major part of the lap. Fuel No. 6 was on equal terms with Test fuel No 5 on the fast track, probably because its higher octane was better managing combustion as piston and combustion chamber temperatures climbed on the two very long uphill straights. However on a shorter circuit similar to those on which of the series was to be fought out, lap times were slower.

Chapter 7

Combustion and The Ignition System

W[illegible] invariably revolves around such things as the size of the engine, cylinder head [illegible]

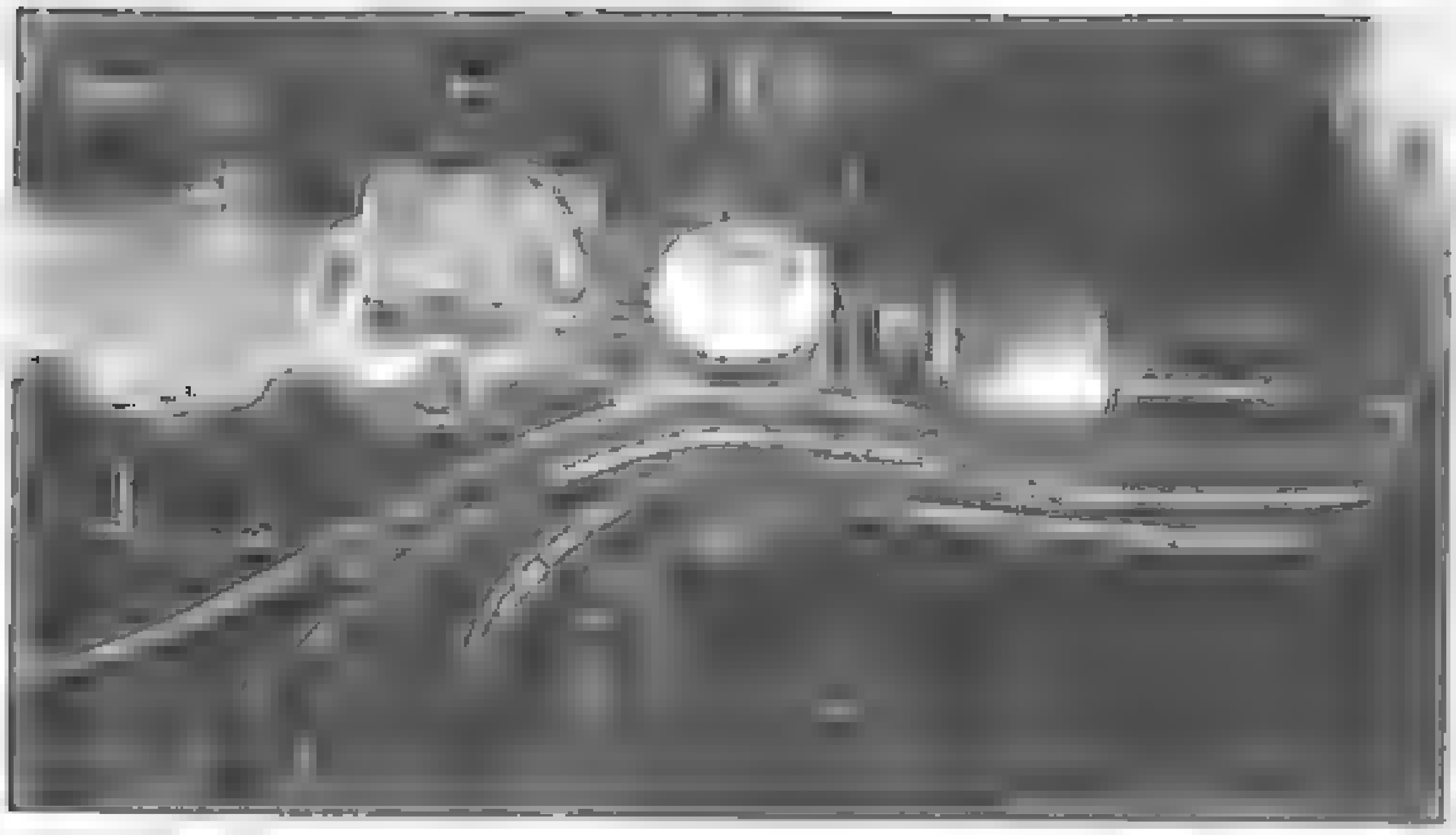
[illegible] easy to come to an understanding and give numbers to the things to which the [illegible]

[illegible] enthusiasts and club racers, nor elevated to its rightful position of importance.

Most of the people involved in engine development appear to understand the [illegible] in getting it into the cylinders in the first place. Consequently if we are to have our [illegible] the correct time with a spark of sufficient intensity to initiate combustion. Then we [illegible] through the entire combustion space at the correct speed. If flame travel is slow, cylinder pressure will rise too slowly to give the crankshaft a good sustained shove [illegible] like a nuclear reactor meltdown eventually destroys the engine.

FACTORS INFLUENCING THE COMBUSTION PROCESS

From earlier chapters you will have gathered that there is a lot more involved with combustion than the ignition system. In reality the ignition system's true function is to



Manufacturers of aftermarket ignition components would prefer us believe that combustion involves nothing more than ... ual ignition system hardware. The reali... is that these parts only start the fir...

initiate combustion. To that end it must be robust enough to get the flame s... today ... e earlier times when some mechanical arrangement ... electronic engine man... ment system contro... hen the spark plug fires to sta... flame off. However there ... e a whole host of f... tors which influence the manner and speed at which that flam... then proceeds thro... the whole combustion space. When we come to understand these factors we are in a much better position to extract th... maximum potential from our engine

The major factors are mixture quality, mixture ... ovement or turbulence within the combustion space, and the design of th... combustion space. These appear to be relatively simple concepts, but unfortunately e... h is extremely complex due to th... that internal combustion en... broad rang... of rpm an... conditions. Therefore what is optimal under one set of conditions will be less th... optimal at other rpm and thrott...

MIXTURE QUALITY AND TURBULENCE

Mixture quality involves filling the cylinder with fuel particles of the corr... evenly dispersed throughout the air. During the preparatory stage prior to combust... we have to break the liquid f... reas... its surface area to th... maximum. That means that outsid... inder we have to choose the correct typ... fuel and fuel in... that fuel into the airstream at t... optimal location and angle. Following that ... to rely on the physical shape of t...

... fairly poorly blended mixture of fuel and air as it enters the cyl...

Once in the cylinder we want this ... ction to continue. The shape of the co...

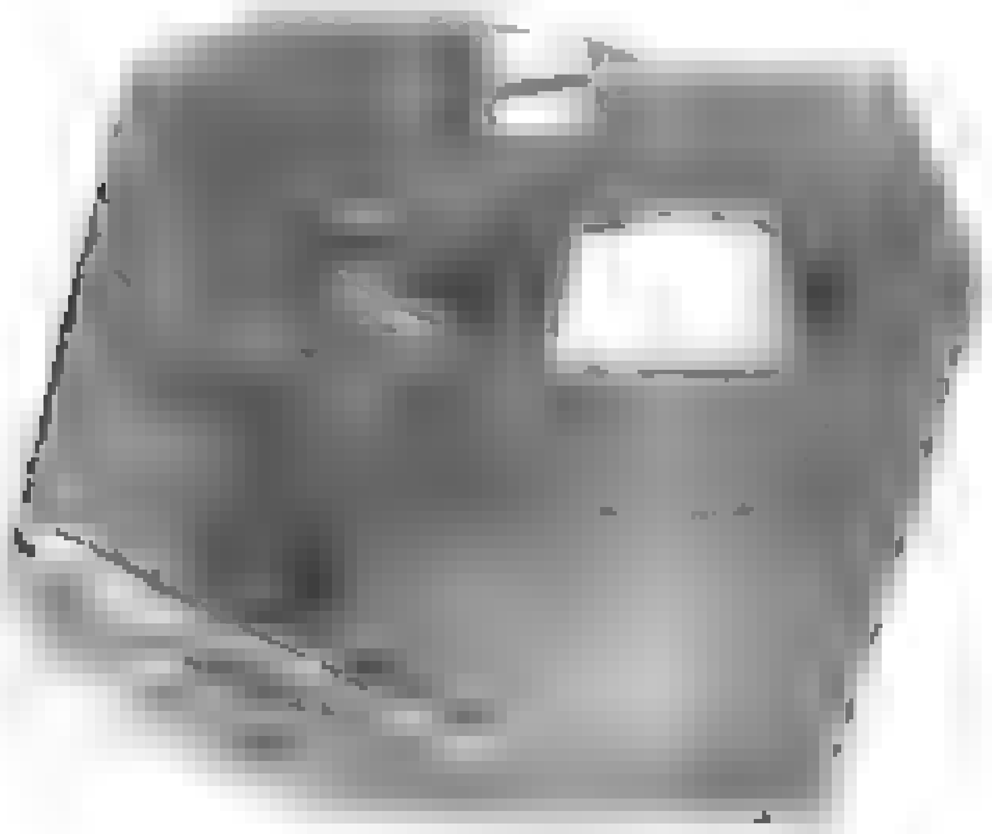
chamber and piston crown then become a major controlling influence. Four-valve combustion chambers and flat top or dish top pistons are superior in this regard as they do not impede the tumbling or swirling action of the mixture. All of us have observed the swirls and eddies in a flooded river, and no doubt you have marvelled at how peaceful the flow can be in one area while right alongside being so turbulent. A similar action is in progress in many two-valve combustion chambers, and also in those engines with high-top pistons. Even very minor redesigns of shallow valve cutouts in the crown of flat-top pistons fitted in 4-valve engines has resulted in power increase in the range of 2–3%. Therefore the impact of the shape of the combustion space on mixture motion and combustion flame propagation should not be underestimated as the possible gains are much greater in engines with less favourable designs.

The purpose of this mixture motion is often said to be to homogenise the mixture – to have an even distribution of fuel molecules and air throughout the cylinder. However this agitation also serves to further break down the fuel into smaller more readily combustible particles. An equally important aspect, and at times an even more important consideration, is the fact that this motion tends to even out the temperature of the mixture. Cool pockets tend to burn slowly and so contribute nothing to power production. The real danger though are hot pockets. These may self ignite the main body of mixture before the spark plug fires. This is what is termed pre-ignition. Another more common scenario is for a hot pocket to self ignite after the spark plug fires. This flame front and the one initiated by the spark plug collide and set off violent combustion of the remaining fuel charge. This is called detonation.

THE SHAPE OF THE COMBUSTION SPACE

We have already touched on how the shape of the combustion space influences mixture motion prior to ignition. Following ignition this area continues to exert its influence. Its design can help the combustion flame spread through the moving fuel mixture at the desired speed. Additionally in the area furthest away from the spark plug the combustion chamber will be pulling some heat out of the mixture, so stalling ignition in these areas where the pressure is rapidly rising, until they are reached by the spark plug initiated flame.

Apart from causing mechanical problems the high top piston seriously disrupts the combustion process



Radial 4-valve piston is almost flat, but minor work to smooth valve cut-outs and modify the crown shape has given good power gains

OTHER ASPECTS OF MIXTURE MOTION

These matters are always a compromise. Thus a road engine

POINTS-TYPE IGNITION

distribute the spark, and a coil to provide the spark (Figure 7.1)

Figure 7.1 Conventional negative earth ignition system

are closed, electric current flows through the coil primary winding, then through the points to earth. [illegible]
surrounds the secondary or high tension winding. As soon as the points open, current [illegible]
volts) capable of jumping across the spark plug gap, to fire the fuel mixture.

The high-voltage current flows from the coil to the centre of the distributor cap and through the carbon brush to the rotor button. The turning rotor then directs the current back to the distributor cap and to the individual cylinders.

At normal engine speeds and low-compression pressures, this conventional system works very reliably, with just periodic maintenance required to replace the points. However, when an engine is modified the ignition system has to work much more efficiently under adverse conditions.

For example, the very first problem we see with points-type ignition is that as [illegible] decreased. This means that the coil has less time to build up a full magnetic field between each plug firing, so the ignition energy available to fire the plugs is reduced and a high-speed misfire occurs.

HIGH ENERGY ELECTRONIC IGNITION

The common electronic transistorised ignition is similar to the contact breaker system in that it is also an inductive storage-type ignition system. However, rather than using mechanical contact points to open and close the low voltage circuit it relies instead on magnetic pulses and electronic circuitry to perform this function (Figure 7.2).

A number of different designs are in use, but the basic mode of operation is similar. The distributor shaft turns a pulse generator rotor inside a stationary permanent magnet. Functioning like a mini alternator, this induces a signal in the pick-

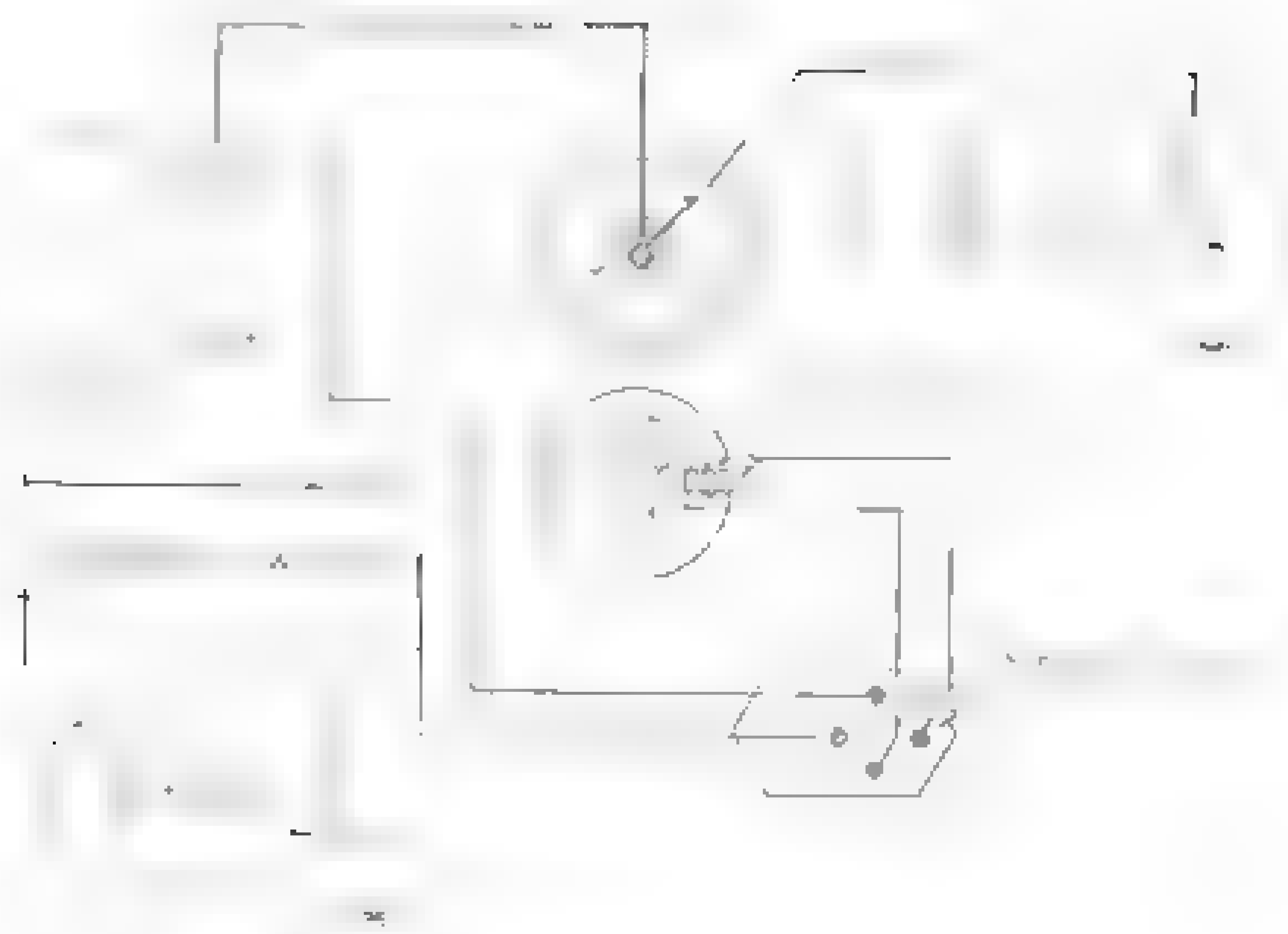


Figure 7.2 Transistor HEI ignition

up coil located within the distributor. This electric signal flows through the electronic control module (also called the ignition module, amplifier or igniter) causing the electronic circuit to switch 'on' or 'off'. When the circuit is switched on, current flows through the primary windings of the ignition coil. When it is switched off, the magnetic field collapses, inducing a high voltage in the coil's secondary winding. As in the points-type system, the high-voltage current then flows to the centre of the distributor cap, through the carbon brush and rotor button out to the individual spark plugs. Because the transistor ignition system does not contain tungsten contact points that pit and burn, a substantially higher primary circuit current and voltage can be utilised to saturate the coil fully, even in the short time available at higher engine speed. Thus a high-intensity, or high-energy, spark is provided, which has led some manufacturers to call this 'high-energy ignition', or HEI.

[illegible] the 1980s for their base domestic engines is a good example of just how well a good stock electronic system can perform in a competition environment. These units were designed to fire across huge plug gaps of up to 0.065in when new spark plugs were first fitted, in a combustion chamber filled with poorly atomised fuel from a carburettor, that was contaminated by lots of exhaust gas intended to keep exhaust emission levels down. Obviously to start a fire and keep it burning, with a gale blowing in the combustion area, the ignition system had to produce a spark that was not only intense, but also of very long duration to put considerable heat into the fuel/air mix close to the spark plug tip. As indicated in Table 7.1 when compared with CD ignition, current output is only 40%, but at 8,000rpm on a V8 the Bosch HEI had a spark duration of 0.4 millisecond, meaning that there was a continuous flame between the spark plug electrodes for 19.2° of crank rotation. So long, in fact, that the ignition module cuts it short to keep the entire spark duration within the combustion cycle, and give the coil adequate time to be recharged with energy to fire the next spark plug.

If you look carefully at the distributor in this big block 440 Mopar, you will see the word 'ICE'. IC & E in Melbourne, rebuild the Bosch HEI distributor, used by Ford and General Motors on their Australian models during the 1980s, to fit a large variety of engines producing up to 1000hp. Because it was designed to get a fire started across huge plug gaps of up to 0.065in, in a combustion chamber filled with poorly atomised fuel from a carburettor and contaminated by big quantities of exhaust gas intended to keep emissions down, the Bosch HEI ignition had to produce an intense, very long duration spark.

Table 7.1 Ignition characteristics – V8 engine test

	Engine speed (rpm)							
	1,000	2,000	3,000	4,000	5,000	6,000	7,000	8,000
Bosch HEI								
Output m amps	50	55	60	60	60	60	60	60
Duration m sec	1.2	1.2	1.2	0.9	0.75	0.6	0.5	0.4
crank°	7.2	14.4	21.6	21.6	22.5	21.6	21	19.2
Capacitor Discharge (multi-spark)								
Output m amps	150	150	150	150	150	150	150	150
Duration m sec	0.13	0.13	0.13	0.13	0.13	0.13	0.13	0.13
crank°	0.8	1.6	2.3	3.1	3.9	4.7	5.5	6.2
Twin-point Conventional*								
Output m amps	30	25	20	15	10	10	7.5	
Duration m sec	1.1	0.8	0.7	0.5	0.35	0.3		
crank°	6.6	9.6	12.6	12	10.5	10.5		
Single-point Conventional*								
Output m amps	30	20	10	7.5	5	2.5		
Duration m sec	1.0	0.7	0.5	0.3	0.18	0.11		
crank°	6	8.4	9	7.2	5.4	3.6		

* Competition points with lightweight contacts and high spring tension to increase rpm at which point bounce set in

[illegible] when comparing the Bosch HEI keep in mind that its output is superior to many stock un[illegible] equal to the best competition HEI systems.

Whereas a points-type system can supply about 18,000 sparks per minute, which corresponds to 9,000rpm for a four-cylinder engine, but only 6,000rpm for a six-cylinder and 4,500 for a V8, a stock HEI ignition raises this figure to just over 20,000 strong sparks per minute, and a good competition HEI system should increase this to at least 30,000. This latter figure means that a top-of-the-line race transistor ignition system should enable a good ignition burn with a single coil at engine speeds of 15,000, 10,000 and 7,500rpm in four-, six- and eight-cylinder engines respectively.

MULTI-COIL AND CAPACITOR DISCHARGE IGNITION

However, what if we require the engine to run at higher speeds, or what if we are burning a fuel that requires a very high intensity spark of long duration to effect good combustion? Depending on race regulations we could opt for an HEI multi-coil system, or capacitor discharge (CD) ignition, with either a single ignition coil or multiple coils. With a multi-coil system more time is available to saturate each coil fully because usually individual coils may be firing only one or two spark plugs, although in some V8 and V12 applications each coil may be firing three or four.

When race rules bar the use of multi-coil systems or we wish to avoid such complexity, we can choose capacitor discharge ignition. Unlike the previously discussed inductive storage ignition systems, CD ignition does not store ignition energy in the ignition coil, but rather in a capacitor. Even the best transistor ignition amplifier and coil combination begins to run out of ignition at about 7,500rpm on a V8. At higher speeds a single-coil HEI system runs out of time for the amplifier to 'soak' the coil, discharge all the energy from it to fire the plug, then fully saturate the coil in readiness for the next plug firing. Thus above 7,500rpm, in single-coil V8 applications, race HEI systems yield less and less secondary spark voltage and energy, the end result being a more and more feeble spark and reduced hp at best, or a misfire at worst.

Some tuners, however, believe that CD ignition is inherently superior to transistorised HEI, and it is therefore the logical choice for a competition engine, even if the engine is running at speeds below that at which a transistor HEI system is running out of ignition energy. The reasoning is that a bigger spark must produce a superior burn, therefore it has to be good for power. However, my testing indicates that an excess of ignition energy does not mean that an engine will produce more power. Only when an ignition system is inadequate will a switch to a better system allow the hp to increase.

Table 7.2 shows what I typically find when a good single-coil transistor ignition is tested against a CD system at engine speeds within the capabilities of transistorised HEI type systems. This V8 Ford Windsor 351 was running in a class that required the use of unported cast iron heads (N351 heads fitted), flat top pistons (12.2:1 compression ratio), and Avgas 100/130 petrol. Clearly at V8 engine speeds below about 8,000rpm one type of ignition does not show any superiority over the other when burning petrol. Also, opening up the plug gaps did not show any change in hp.

CD ignitions in single-coil V8 applications should not begin to run out of ignition energy to speeds in excess of 10,000rpm because it takes considerably less time to charge and discharge a capacitor than it does to charge and discharge a coil. In

 CD systems current in the primary circuit powers a mini-oscillator/transformer, which

Table 7.2 Ford 351W ignition system comparison

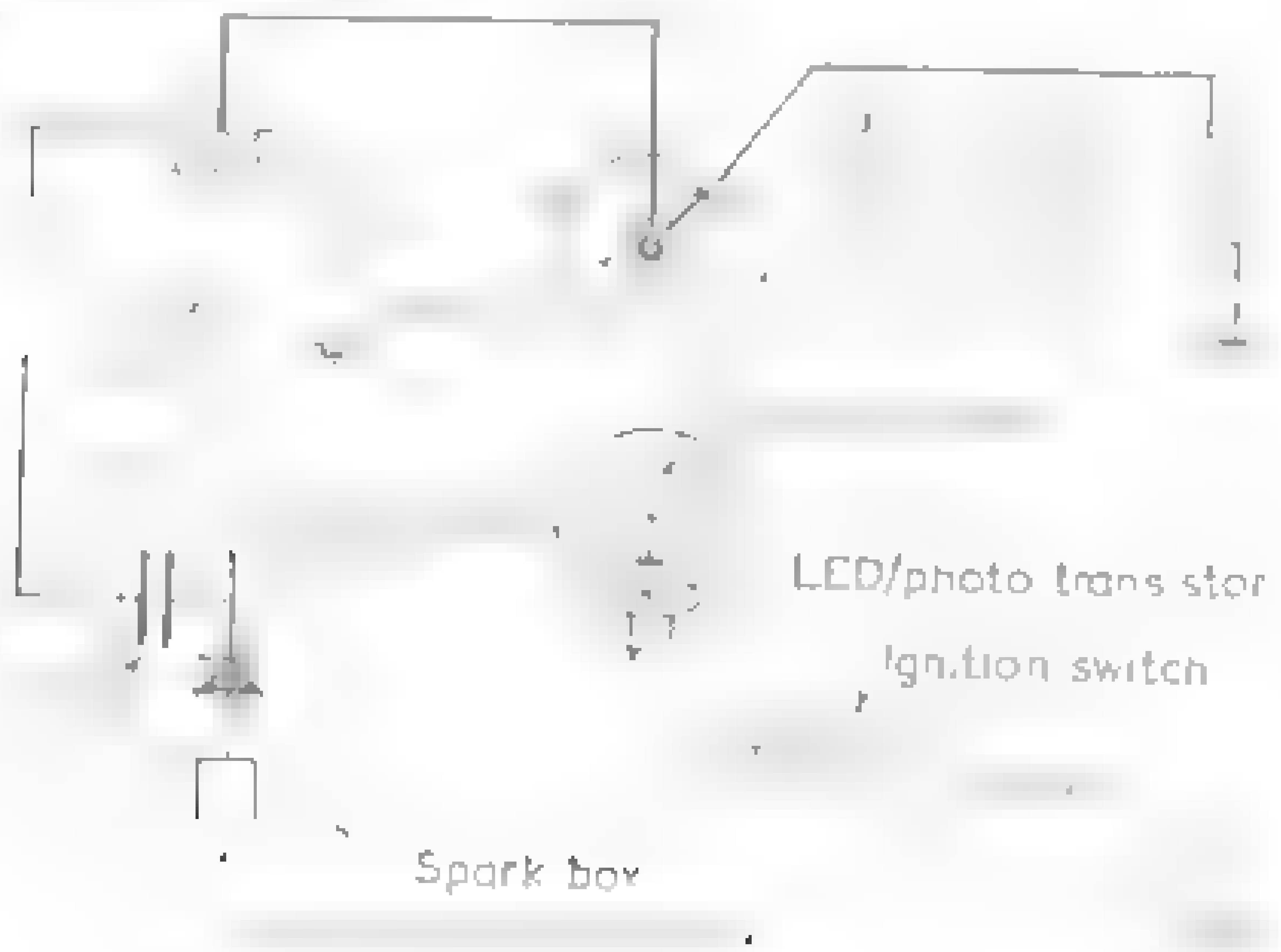
rpm	Test 1		Test 2		Test 3	
	hp	Torque	hp	Torque	hp	Torque
4,500	386	451	388	453	385	449
5,000	430	452	431	453	430	452
5,500	467	446	466	445	466	445
6,000	502	439	500	438	500	438
6,500	530	428	531	429	530	428
7,000	542	407	544	408	544	408
7,500	540	378	538	377	538	377
8,000	436	286	440	289	439	288

Test 1 – Ford SVO transistorised high energy ignition, 0.040in plug gaps

Test 2 – Allison (now Crane) capacitor discharge ignition, 0.040in plug gaps

Test 3 – as above but with 0.060in plug gaps.

and plug leads in extremely quick time (Figure 7.3)



To enable the Cosworth DFV Formula 1 engine to work at over 10,000rpm Lucas developed this CD ignition unit

Rise time for a conventional ignition is 75–125 microseconds, but CD systems have a rise time of only 20 microseconds. This enables a CD ignition to fire badly fouled or wet plugs, which means that an engine that has been 'loaded up' during a spin and stalled will quickly fire again. With other systems the rise time may be so slow that a wet plug will bleed off voltage across the fouled insulator, and there will be insufficient ignition energy left to bridge the plug gap and get the engine running. A CD ignition, however, delivers full energy to the spark gap, because the voltage is produced so quickly that it jumps the plug gap before the spark energy has time to bleed off.

MULTI-SPARK CD IGNITION

A disadvantage of ordinary CD ignition is that it produces a sp[illegible] duration. What happens is that there is [illegible] ple ignition energy to bridge the sp[illegible] electrodes, but because the spark is of such short duration, turbulence [illegible] combustion chamber blows out the spark before it gets sufficient heat into the fu[illegible] mixture to initiate a good combustion flam[illegible]. This problem led to the development of multi-spark CD systems, which typi[illegible] multiple sparks for up to [illegible] crank rotation. In turn the introduction of mu[illegible]lti-spark CD ignition has led to a lot of misunderstanding about CD sys[illegible] up [illegible] for [illegible] using propaganda to a[illegible]

First, let us consider the multi-spar[illegible] [illegible]. Some advertising seems [illegible] that the ignition will supply a s[illegible] [illegible] sparks per [illegible]

engine rpm. This is just not so, except at very low engine speeds when the ignition can [illegible]

reverted to just one fat spark that may be all over in less than 3° of crank rotation. [illegible]

140mj of stored spark energy at 9,000rpm.

To further complicate matters manufacturers often do not specify the energy [illegible]

they obtain that figure. It is the stored energy level per spark at high rpm that really counts.

Even then we run up against a problem in that energy losses vary from one [illegible]

more involved than merely looking at raw numbers, as these two systems could [illegible]

race HEI and points-type systems will be hard pressed to deliver better than 50% of their peak stored energy, about 95mj and 28mj respectively, at the spark plug.

IGNITION VOLTAGE REQUIREMENTS

At this point we should consider how an engine's secondary voltage requirements [illegible]

systems that will deliver, say, 50,000 to 60,000 volts in a race engine. This is wrong thinking, because at anything in excess of 40kV in a distributor cap – regardless of whether the system is a distributor or distributorless type – you will see sparks jumping to earth from plug wires and boots. Every time this sort of thing happens there is no fire at the plug gap, so hp falls off.

Really the engine's voltage requirements depend mainly on the width of the spark plug gap, and the compression ratio. At the bottom end of the scale an engine running 0.028in plug gaps and a compression ratio of 11:1 will require about 20kV to get a spark across the plug electrodes. A 0.040in gap and a 12.5:1 compression ratio will push the voltage requirement to about 27kV, while a 0.060in gap and a 15:1 compression will see this requirement rise to around 35kV.

The effect of this secondary voltage requirement in drawing off stored ignition energy is substantial. In the above three examples, with voltage requirements of 20, 27 [illegible]

Table 7.3 Ford 351W ignition system comparison

rpm	Test 1 hp	Test 1 Torque	Test 2 hp	Test 2 Torque	Test 3 hp	Test 3 Torque
4,500	416	485	417	487	418	488
5,000	463	486	466	490	465	488
5,500	511	488	514	491	515	492
6,000	555	486	557	488	560	490
6,500	587	474	593	479	588	475
7,000	601	451	613	460	613	460
7,500	597	418	615	431	614	430
8,000	471	309	490	322	495	325

Test 1 – Ford SVO transistorised high energy ignition, 0.040in plug gaps

Test 2 – Allison (now Crane) capacitor discharge ignition, 0.040in plug gaps

Test 3 – as above but with 0.060in plug gaps

The converse must also be taken into account, ie when race regulations or our budget preclude the use of high-energy ignition systems, we must do something about reducing our engine's ignition energy requirements. The place to start is to reduce the

spark plug gap in line with what is discussed later in this chapter. I know a lot of noise has been made about the [illegible]

[illegible]

Table 7.4 Spark plug gap comparison with stock HEI system

rpm	Test 1		Test 2		Test 3	
	hp	Torque	hp	Torque	hp	Torque
3,500	246	369	252	378	245	368
4,000	327	429	330	433	324	426
4,500	412	481	411	480	410	478
5,000	455	478	459	482	456	479
5,500	489	467	487	465	493	471
6,000	498	436	491	430	500	438
6,500	489	395	478	386	494	399
7,000	441	331	423	317	445	334

Test 1 – 0.035in spark plug gap

Test 2 – 0.045in spark plug gap

Test 3 – 0.030in spark plug gap.

CHOOSING AN IGNITION

As you have already seen from the example of the Bosch HEI introduced over 25 years [illegible] the factory ignition isn't up to the task. The aftermarket ignition manufacturers might want you to believe the stock [illegible]

[illegible] extremely wide spark plug gaps. The main area where stock systems have a problem is at very high rpm. However you can reliably assume that as you will invariably be operating on considerably richer mixtures and with smaller spark plug gaps the power differences

between the stock system and high performance ignitions will be negligible up to the rev limit set by the car manufacturer. In fact I've seen many aftermarket systems give marginally less power and a few gave a lot less at such rpm. When the revs rose way beyond the factory rev limit though, or when a rich methanol mixture had to be ignited, then some aftermarket ignitions significantly outperformed the stock unit

Another important factor in choosing an ignition system is its reliability Therefore ask the manufacturer about their standard 'burn-in' and [illegible] or black box) contains numerous electronic components, which, if they are going to fail, will do so in the first few hours of operation. With this in mind we should be looking for a unit from a manufacturer where every spark box coming from the factory is given a 4–6 hour burn-in and testing with normal ignition loads before it is packaged for sale. Obviously an amplifier subjected to this type of testing may cost a good deal more than if the manufacturer were to pull one box off the production line out of every hundred, or a random unit every couple of weeks, for a burn-in and test

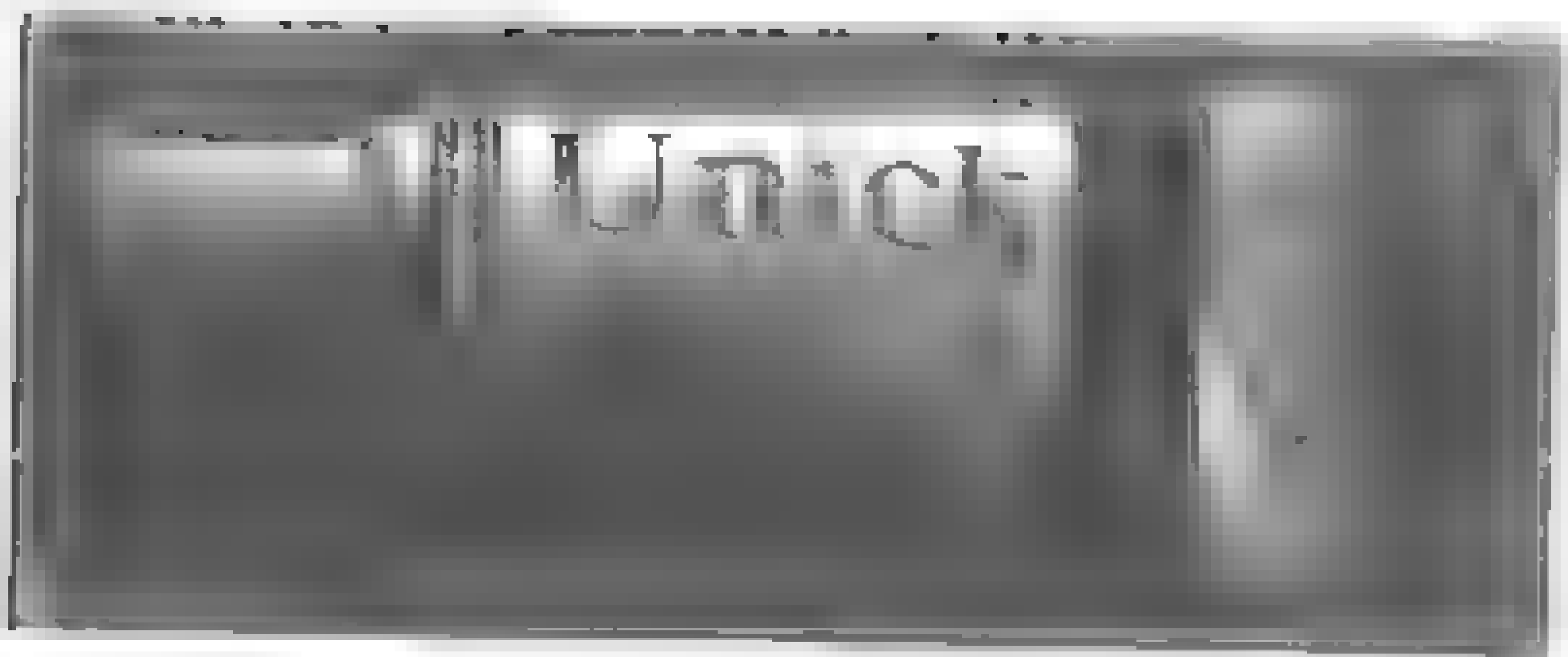
The other concern is amplifier repairability Some manufacturers, after mounting all of the electronic components, fill the spark box with either a rubbery or a hard epoxy compound to keep out air and moisture. However, when the amplifier breaks down all this filler material makes it virtually impossible to replace failed components, so the entire ignition box has to be scrapped The alternative sealing method used by some other manufacturers involves dipping the mounted amplifier components, but the case is not filled with any epoxy When any parts break down the sealer can be penetrated, the defective part replaced, then the assembled components all dipped once more to reseal the electronics

HELPING THE IGNITION LIVE

However, having said all that, most high-energy ignition troubles are caused by the user, not the manufacturer. The basics to keep in mind are that these systems will not tolerate excessive heat, vibration or moisture, and they quickly fail from high voltage build up within the system. Therefore before fitting the ignition into the car read the installation instructions carefully. If you are not clear on some point, talk to the manufacturer. On the other hand, if the instructions appear to make sense after a quick read through, re-read them, no instructions are that easy to follow, and you may have misread something!

Be sure to mount the spark amplifier and the coil away from the headers, and shield them from heat being radiated or blown off the headers. If it is practical direct a small cool air duct over the spark box. Whenever components mounted on a heat sink are replaced, take care to apply the appropriate heat sink grease when the new components are being fitted. To control vibration use the mounting rubbers provided by the manufacturer, and if the rubber pads are through bolted do not squash them tighter than specified

Do not blast any part of the ignition system with a pressure washer. If any sealer or insulator is less than perfect a tiny fraction of moisture could see a high-voltage flash-over and subsequent system failure when the engine is started – even after several days' drying time. Keep in mind also that detergent and degreaser residues are corrosive and conduct electricity



Heat and water are enemies of electronic components. Therefore whenever components mounted on a heat sink are replaced, be sure to help heat transfer by using the appropriate heat sink grease

To avoid power overload in any part of the system, never disconnect any wires with the engine running. In similar vein, never crank the engine over with any of the spark plug leads or the coil lead disconnected, without first isolating the ignition system. With many systems the engine can be cranked while the leads are disconnected providing that the low-voltage positive (+) ignition wire is disconnected at the coil, but check this with the manufacturer.

Battery chargers and welders have also taken their fair toll in destroying ignition components. When using a battery charger with the battery still fitted in the car, disconnect both battery cables before hooking up the charger. When welding on the car disconnect both battery cables and disconnect the main harness going to the spark box as well as any earth straps connected to it. Also put the welder earth clamp in a place where it can make a good ground connection close to where you want to weld.

IGNITION ADVANCE REQUIREMENTS

With any type of road ignition system and most types of competition ignition, some arrangement is employed to adjust the ignition timing angle to suit various engine operating conditions. For example, at idle the ignition may be timed to fire the plugs at anything from 5° to 20° before the piston reaches top dead centre (TDC) on the compression stroke. However, at wide-open throttle and at high engine rpm the spark lead needed for best hp may be around 25° to 35° BTDC for engines burning petrol (on alcohol or nitro add about 10° and 20° respectively). Then at cruise on highway, or under a yellow light at the speedway, an additional 10° to 15° advance may be necessary to give best economy and performance.

These changing ignition advance angles are necessary to give the fuel/air mixture the correct amount of time to burn properly. Most engines produce maximum power when we start ignition so as to reach peak cylinder pressure at around 12–14° after TDC. Clearly at 7,000rpm the combustion flame must have been progressed the same amount as it would have at 13,000rpm, so that maximum energy is available to force the piston down and produce power at the crankshaft. Remembering that most power-producing force has been expended by mid stroke, it should be obvious that the only way that this deficit can be made up is to start the burning of the combustion flame earlier.

Apart from the time factor, there are several other reasons why the spark angle must be varied for different engine operating modes, and why the ignition advance

changes from one engine to another. The main contributory factors are the fuel/air ratio, mixture density, exhaust gas reversion, engine design and fuel type.

Both very lean and rich fuel/air mixtures burn slowly and require more spark lead. A mixture close to full power lean burns the fastest and requires less advance. At cruise, for example, road engines are usually tuned to be a little lean in the interests of economy, so more advance is necessary. However, mixture density is also a factor, because in this situation, with the throttle just cracked open, the cylinders do not become crammed with fuel/air mixture, so even when compressed as the piston rises to TDC, the oxygen and fuel molecules are separated slightly, which in turn slows the speed at which the combustion flame travels through the combustion chamber.

Understanding this you will appreciate that an increase in the compression ratio has the reverse effect. This increases mixture density, so we have to reduce the spark lead. A cam change also influences mixture density, as do any changes made to the intake or exhaust, which affect the volumetric efficiency of the engine. When cam lobe duration is increased, mixture density will be reduced at lower rpm, while in the mid-range and at maximum rpm it will be greater. To compensate, ignition advance will have to be increased at lower engine speeds, then reduced higher up the rev range. Changes to the intake system have a similar effect on mixture density, but the influence of the exhaust system is different.

When an efficient exhaust system is fitted the amount of exhaust gas left unscavenged in the cylinder, or reverse flowing into it, is reduced. Th
room in the cylinder to pack in fuel/air mixture, so mixture density increases and the spark lead must be reduced. The other effect of upgrading the exhaust system is that there are fewer inert exhaust gas molecules to separate the oxygen and fuel molecules, so flame speed through the combustion space is faster, which again calls for less advance.

Engine design also figures in the amount of ignition advance required for efficient combustion. The size of the combustion space and the position of the spark plug(s) obviously influence the amount of spark lead needed. The further the flame has to travel, as in a large combustion chamber with only one spark plug, the longer it will take to burn the mixture. Conversely, the closer the spark plug is to the centre of the chamber, as in four-valve chambers, the faster the combustion time. Cylinder heads with inlet ports and combustion chambers that impart a high degree of swirl to the inlet charge improve mixture homogenisation, and this speeds up flame propagation so less advance is called for. Increasing the amount of squish, by reducing the squish clearance between the piston and head, has a similar effect. The stroke of the crankshaft, and the ratio of the con rod length to the length of the stroke, can also influence the amount of ignition advance necessary because these two factors affect the time it takes the piston to move to, and just past, TDC.

The fuel compounds used also affect combustion speed. Petrol burns fairly rapidly, so requires less advance than other fuel types. Alcohol fuels burn more slowly, and nitro is slower still, which means that these fuels need more spark advance. However, when a flame accelerator compound is added to either alcohol or nitro, spark lead must be reduced.

Obviously in a modified engine some, or perhaps all, of the above factors will have been changed from the stock engine. Therefore the amount of spark advance needed under various engine operating conditions will probably be quite different from that determined by the factory to achieve best power and economy.

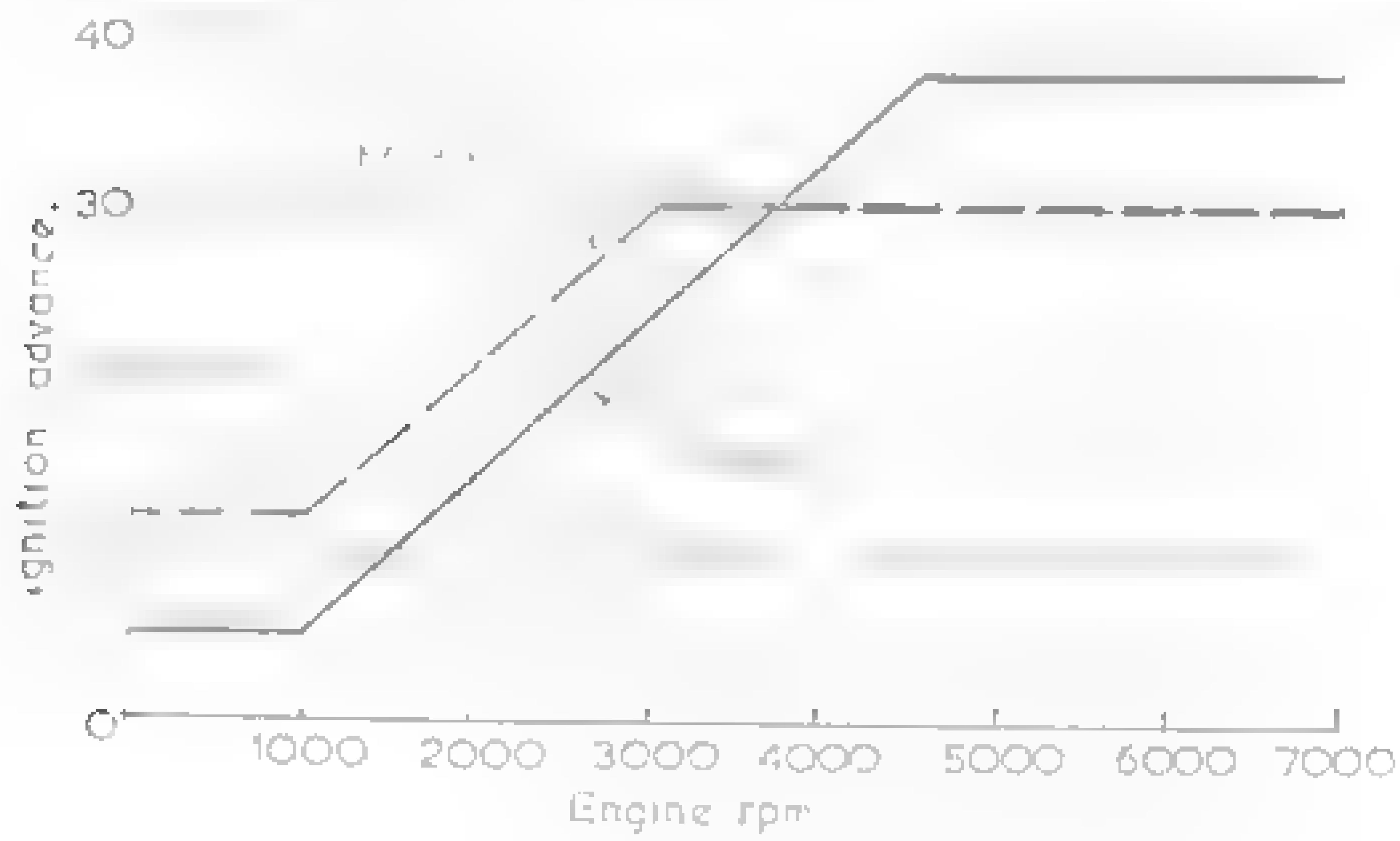
MECHANICAL ADVANCE LIMITATIONS

Points-type distributors and early electronic ignition distributors relied on a mechanical system of revolving bob weights and springs, along with a vacuum [illegible] operating conditions. We will not discuss how these distributors can be modified to [illegible] also mean an engine wrecked by detonation.

Looking at Figure 7.4 you can begin to appreciate the sort of problems that are encountered when we are stuck with a mechanical advance system. In a stock situation [illegible] 4,500rpm, where it reaches 38°. That 38° spark angle is then maintained right up to the engine's rev limit.

As explained previously, when the engine has been modified the advance [illegible] competition engine. Because of the increase in the camshaft lobe duration, the engine now requires a lot more advance early on in the rev range, but with better cylinder filling at higher rpm less ignition lead is required past about 3,000rpm. Thus the initial advance in this example is adjusted to 12° before TDC and the distributor springs and weights are modified so that the distributor advances the timing very quickly to 30° at 3,000rpm. Again note that the spark lead advances in a linear fashion between 1,000 and 3,000rpm, and past 3,000rpm the timing is maintained at 30° even though dyno tests show that this

Figure 7.4 Mechanical distributor advance curves

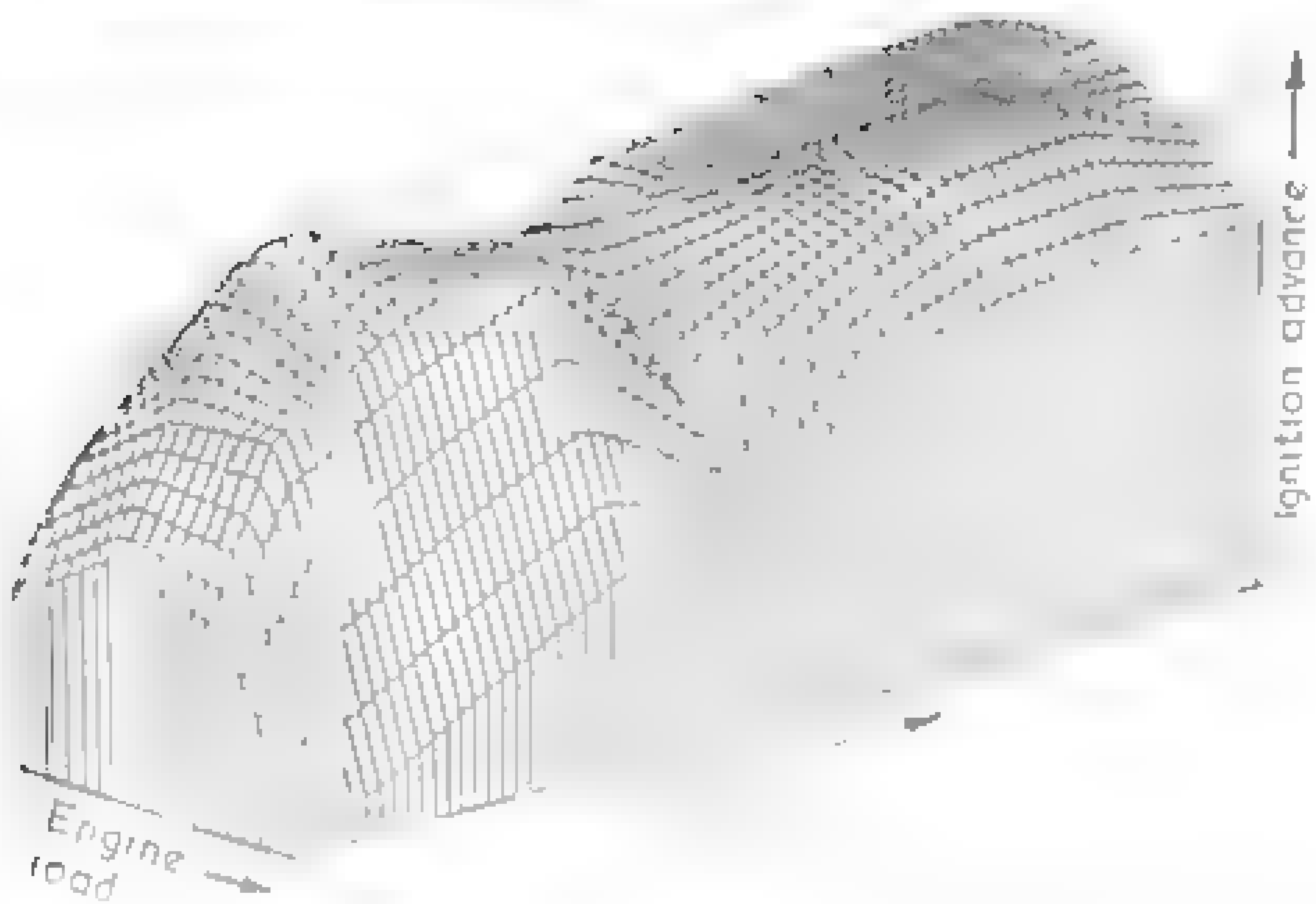


engine would make more power with the advance at 33° at 6,500rpm. Unhappily because of the mechanical nature of the

excessive advance at an engine speed when cylinder filling is superior.

ELECTRONIC ADVANCE CONTROL

However, fully computerised ignition does not constrain our search for maximum power and performance. The 3D



prefers to make best power at wide open throttle. For a competition engine with a 4,000 to 8,500rpm working range this may be, say, 34° at 4,000rpm, 33° at 4,400rpm, 31° at 4,800rpm, 32° at 5,200rpm, 34° at 5,600rpm, 36° at 6,000rpm, 38° at 6,400rpm, 36° at [illegible] Clearly this is the sort of advance curve that no mechanical system could achieve. The advance angles would then be stored in the memory and later, whenever the ECU [illegible] this would change to 36° ignition lead, and so on.

This dyno test process would then be repeated at various throttle plate angles. If [illegible]

Just how much power increase can we realistically expect with computerised [illegible] 1½-2½% with a circuit race engine that breathes very efficiently and has a fairly narrow power band running from, say, 6,000 up to 8,500rpm. An engine tuned for a wider power range running from, say, 4,500 to 8,500rpm, as would be used in a rally car or endurance racer, has more compromised high-rpm breathing. Hence it benefits much more from the additional spark advance because of its poorer cylinder-filling abilities in the upper rev range, so may show around a 4% hp increase.

Competition engines typically want an additional 3 or 4° spark lead at peak hp revs than that required at peak torque rpm. Then, past peak hp, they may want another [illegible] the difference, so if the engine calls for 32° at peak torque rpm and 36° at the rev limit, the tuner will set a compromise angle of 34° and hope that the engine does not detonate and break a piston due to excessive advance at the torque peak. Electronic [illegible] some of that wheelspin by taking off spark advance at the bottom of the power band when we run at that track. If we have enough ECU capacity we can run several

Table 7.5 Lotus Ford Twin Cam ignition advance mechanism comparison

	Test 1		Test 2	
rpm	bhp	Torque	bhp	Torque
4,000	89.9	118.1	89.6	117.6
4,500	104.0	121.4	104.3	121.7
5,000	121.6	127.7	119.4	125.4
5,500	138.0	131.8	137.7	131.5
6,000	154.9	135.6	155.1	135.8
6,500	159.8	129.1	159.0	128.5
7,000	165.8	124.4	165.5	124.2
7,500	171.2	119.9	168.4	117.9
8,000	172.4	113.2	168.5	110.6
8,500	166.2	102.7	160.7	99.3

Test 1 – Allison ignition with electronic advance control

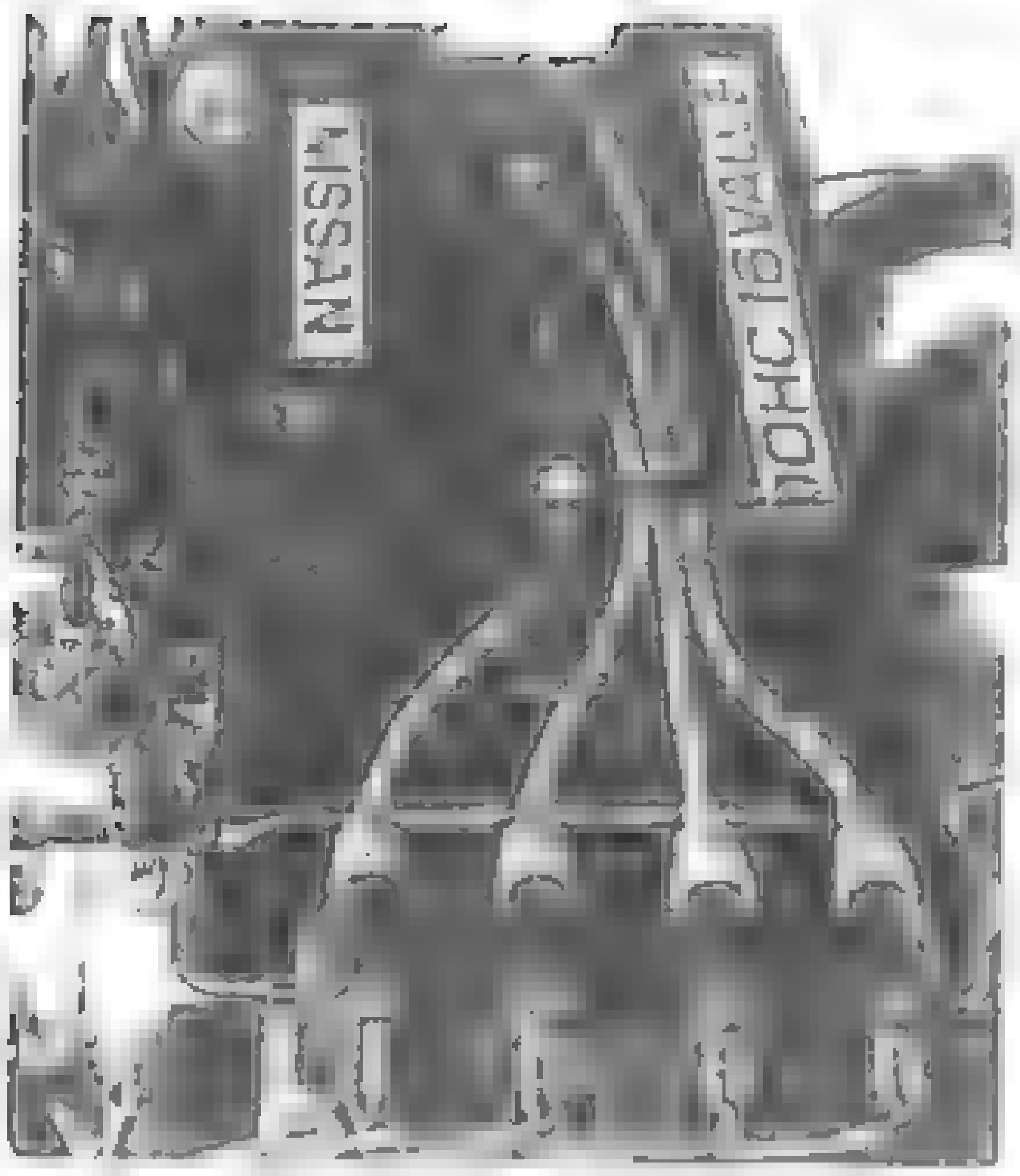
Test 2 – Allison ignition with mechanical advance control

advance programmes to help with traction. In the basic system we can run one spark curve that kills mid range torque when in first, second or third gear. Then in fourth, fifth or sixth gear the selector sensor switches to the normal advance curve. A more advanced system would have a gear selector sensor and also a dash control knob with three or perhaps four possible positions. On a circuit racer those positions would be marked 'dry', 'damp' 'wet' and 'yellow'. The first three positions allow the selection of an advance curve to suit track surface conditions, while the 'yellow' position would not normally call up a different advance curve but would rather 'trim' a certain percentage off the fuel injection pulse width when running under a yellow flag. This would lean the fuel right off so as to avoid the risk of fouling the plugs, and in a distance racer it would cut fuel consumption if the yellow was on for a lengthy period.

IMPROVING SPARK TIMING ACCURACY

When we switch to electronic advance control we no longer need a distributor with [illegible] or else by a jackshaft driven by the timing chain. Either way, harmonics in the valve train play havoc with the motion of the distributor and the problem worsens as valve spring pressures and engine rpm increase.

Added to this, some engines have the cam drive at one end of the camshaft and the distributor drive at the opposite end. This does not present a problem at ordinary engine speeds, cam lobe lifts and valve spring pressures, but in a competition engine the story changes dramatically. Engine speeds rise spectacularly, cam lobe lifts rise and the camshaft diameter may decrease to achieve such a big lift, and valve spring pressure may double or treble. What happens now – particularly in push rod V8 engines – is that the cam tends to screw up, with the distributor drive end of the shaft lagging behind the other end where the timing sprocket is attached.



Multi coil distributorless ignition eliminates problems with timing accuracy and distributor cap flashover, plus it extends the rpm potential of transistorised HEI ignition

However, the distributor end of the cam does not trail behind for ever; depending on which valves the cam is pushing open, the camshaft will wind up so far, then instead of the distributor end trailing, it will unwind and spring forward. This means that the triggering signal being intercepted by the ECU to reference TDC, and thus determine the spark firing angle, is unreliable and inaccurate. In reality this means that what should have been a 32° before-TDC firing angle at 8,000rpm may become a 35° angle on cylinder No 1, a 29° angle on No 2, 33° on No 3 and 32° on No 4. Obviously, because in this example No 1 is firing 3° advanced due to camshaft 'spring', we have to retard the timing on all of the other cylinders so as to avoid detonation in the No 1 hole. This costs power and it may cost us in another way. Those cylinders that are now retarded may 'afterburn' due to late ignition. In turbo engines this could reduce turbo life and limit our boost pressure. In naturally aspirated engines the temperature of exhaust valves and valve seats could up over the edge and introduce reliability problems.

Distributors driven from the camshaft by a helical gear experience timing error problems in a different manner. Unless the distributor shaft is shimmed to keep end play to a minimum – 0.003in is preferred – the distributor gear will ride up and down on the cam gear and upset timing accuracy. Likewise a camshaft 'walking' backwards and forward in the block due to excessive end float also contributes to erratic ignition timing. Cam end float should be 0.003in, controlled either by a thrust plate or, where one is not fitted by the factory, a timing cover incorporating a thrust button and [illegible] reinforcement, to keep cam walk to a minimum. However, even when both distributor shaft end float and cam walk are kept to the specified minimum, a measu

When we get rid of the distributor we also rid the engine of another potential trouble spot, namely the distributor cap. With small spark plug gaps and low secondary ignition voltages, ignition flashover, or crossfiring, within the cap is rare. However, the situation changes as plug gaps are opened up and ignition voltages increased. With four- and six-cylinder engines it is possible, by going to a large cap diameter, to put more space, and hence a bigger air gap, between the terminal posts. However, with eight-cylinder engines it can be difficult to find sufficient space to fit such a large distributor cap, and the problem is compounded when we go to 10- and 12-cylinder engines, as the cap diameter grows ever larger.

If we have enough ECU capacity, accurate distributorless ignition also opens up the prospect for ignition with individual cylinder trim, which means that we can adjust

Correcting the distributor

[illegible]

the spark firing angle to suit the characteristics of each cylinder. In this way each cylinder can be tuned as if it was an individual engine. For example, due to engine breathing characteristics, less than perfect coolant flow through the block and head, or [illegible] may run hotter or cooler than others. With ignition trim we can get more power out of the cool-running cylinders by giving them more spark advance. Additionally, we may get more power, or at least better reliability, when we cut back on the advance angle for the cylinders that are running hot.

[illegible] flywheel. Regardless of their position, as each firing pin moves past the stationary [illegible] advance angle and switches the ignition primary (low-voltage) circuit on and off.

Probably the main things to watch with these crank trigger-type systems are that [illegible] gap to the magnetic pick-up is correctly adjusted. The need to have the timing pins [illegible] the minimum pin diameter, but in any case it is a good idea to use a diameter not much smaller than the size of the pick-up. Thus if the pick-up is about 6mm in diameter, use firing pins with a 6mm head. If you use 3mm pins the pick-up may not 'read' them accurately, leading to all kinds of problems. Likewise the air gap where the firing pin passes the pick-up must be adjusted to that recommended by the manufacturer, usually 0.020–0.040in, otherwise you will have trouble getting the engine to fire up.

DETERMINING OPTIMUM IGNITION TIMING

Regardless of the type of ignition system used, it is imperative to get the ignition timing spot on for your engine. Many enthusiasts have a tendency to over-advance the timing in an effort to pick up every last fraction of performance, but my advice is to use the least amount of advance conducive with peak performance. Years ago the advice I regularly heard was to bump up the timing until the engine rattled, then back off the advance 2°. This sort of tuning may get you close with a low-performance, close-to-stock, road engine, but it is not good advice for a competition engine. You do not hear a race engine rattle, and if you do, the damage is already done.

The place to get the timing sorted is on the dyno. Keep adding spark lead while watching the read-out, and when adding additional advance brings no power increase, or even perhaps a drop in hp, back off the timing, as you have gone too far. If the timing is adjusted using a transient 'acceleration' dyno test rather than a 'steady state' step test, the timing could be way too far advanced, depending on the acceleration rate programmed into the dyno. This occurs because the load is not held for sufficient time to get a lot of heat into the combustion chambers and pistons. Therefore the engine will take more advance before it detonates or the power falls off than it would in the real world. Acceleration tests are useful for initial quick setting up, but keep the rate down to 100rpm/sec, and back up the results with steady state dyno pulls.

If you run very close to the maximum total advance without going into high speed detonation, you must be very careful to compensate for changes in the atmospheric conditions. A 7–10% increase in the relative air density (RAD) could result in a fuel air mixture lean enough to produce a set of melted pistons, if you do not enrich the mixture accordingly. Also a decrease in the relative humidity could see the engine detonating unless the spark advance, or the amount of cam advance, is reduced.

ABNORMAL COMBUSTION

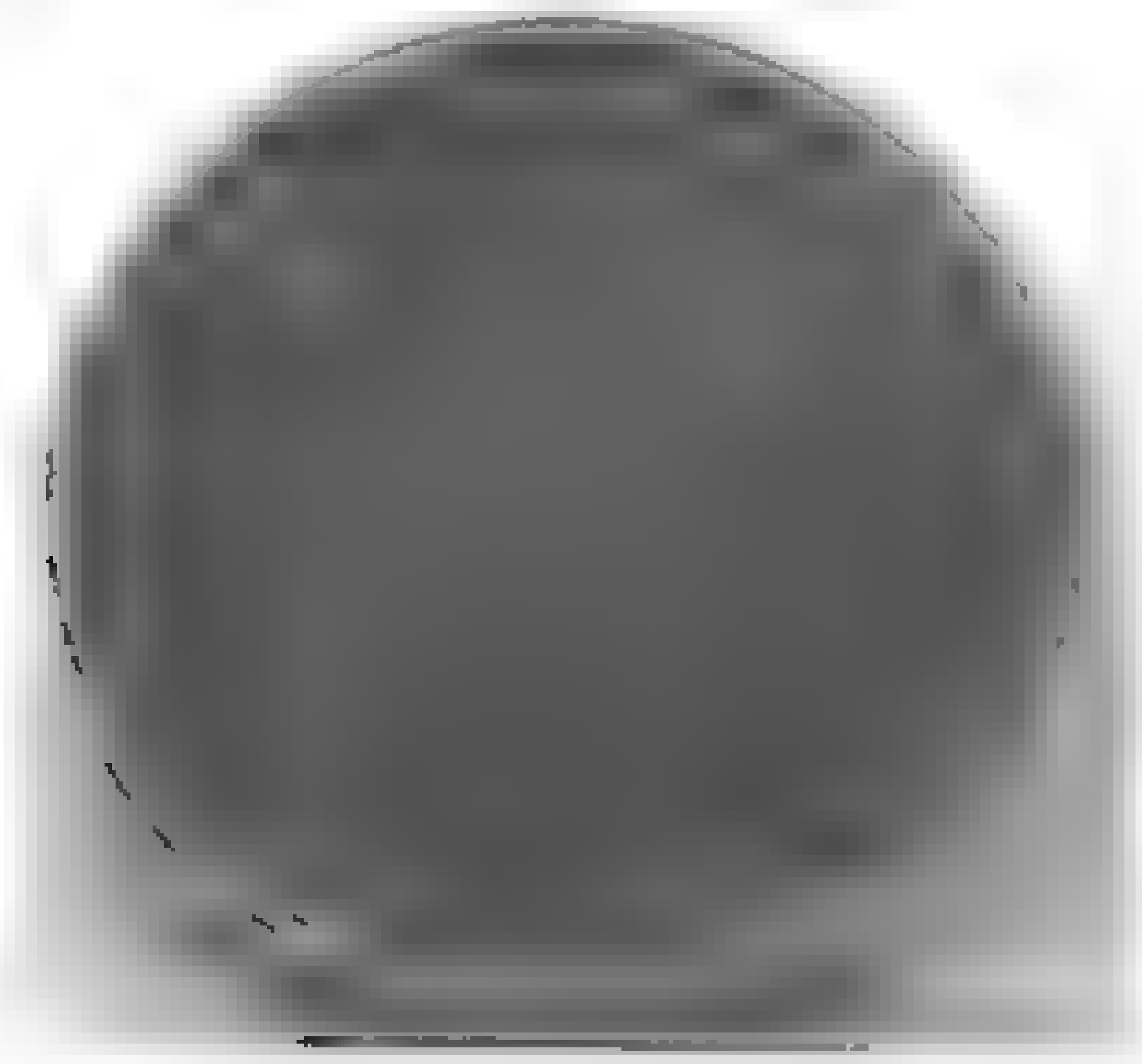
Detonation and pre-ignition are both engine wreckers. Pre-ignition is self-ignition of the fuel caused by a hot spot within the combustion chamber, or due to the fuel becoming unstable because of excessive pressure or heat. Detonation is a violent burning of the fuel (almost an explosion) caused by colliding flame fronts after the spark plug has fired.

When engine damage results from either condition, the culprit can usually be identified after an examination of the pistons and spark plugs.

Pre-ignition damage is caused by extreme combustion temperatures melting the top of the piston and possibly also the ring lands. If a hole is present in the piston crown, it will appear to have been burned through with a welding torch, and the metal around the hole will have a melted appearance. Spark plugs exposed to sustained pre-ignition quite often have the centre electrode melted away, and in extreme cases the insulator nose may also be fused.

Usually, pre-ignition can be traced to combustion chamber or exhaust valve deposits becoming incandescent, but it may also be due to blocked water jackets creating a hot spot, or a glowing spark plug with a heat range too hot for the engine. In a few cases pre-ignition can be traced to an overheated piston, perhaps due to inadequate lubrication, improper clearance or a broken ring.

Piston erosion almost directly under the spark plug or around the edges is a sure indicator of detonation or pre-ignition.



A piston damaged by detonation will show signs of pitting on the crown, in extreme examples the piston crown may be holed. The hole will appear to have been punched through, with radial cracks and a depressed area around it. A spark plug subjected to detonation will usually show signs of cracking at the insulator nose.

When detonation occurs, a portion of the fuel/air charge will begin to burn spontaneously due to excessive heat and pressure after normal ignition takes place. The two flame fronts ultimately collide and the resulting explosion hammers the engine's internal components. Detonation can be attributed to excessive spark advance and/or lean fuel/air mixtures. In supercharged and turbocharged engines, excessive intake charge temperatures can also lead to this condition.

THE ROLE OF THE KNOCK SENSOR

Many modern engines have one or more knock sensors fitted by the manufacturer. In these cars the sensor alerts the ECU of any abnormal combustion condition. The ECU then takes action, depending on how it has been programmed, to bring combustion back to normal. This may be achieved by retarding the spark, adding more fuel or lowering boost. In practice this usually works reasonably effectively with stock and lightly modified engines. However as we get into higher levels of tune the ECU most likely will not be able to bring combustion back on track due to programming limitations. Consequently the engine may be damaged if we leave it to the knock sensor and ECU to control combustion.

Another problem is that when we modify an engine the knock sensor may begin responding to phantom 'noise', assuming it to be abnormal combustion, and then set about reducing hp to 'protect the engine'. In reality those vibrations detected as knock could be due to increased piston or bearing clearances, wilder cams and increased valve train harshness, or even the fact that the engine is now spinning at more rpm. Some stock engines also generate phantom signals from such sources and manufacturers get around the problem by filtering out signals from the knock sensor above a certain number of rpm, or over a certain rev range. Too bad if the engine actually detonates during those periods!

Consequently I prefer to disconnect the knock sensor from the ECU and instead have it activate a warning light and a buzzer to alert me to knock, either real or phantom. Immediately I can lift off the accelerator to protect the engine and then set about adjusting fuelling and ignition advance, and perhaps lowering engine temperature. If the knock light still came on I would then keep running the engine in that condition for a few seconds. Following that the plugs would be pulled and carefully examined, and likewise for the tops of the pistons using a borescope poked down the spark plug holes. If there was no evidence of detonation, the inference would be that engine harmonics or rough combustion at those rpm and throttle opening were setting the knock sensor off.

SPARK PLUG HEAT RANGE

Due to increased combustion temperatures in a modified engine, consideration must be given to find a spark plug with the correct heat range. A hot plug transfers combustion heat slowly and is used to avoid fouling in engines with relatively low

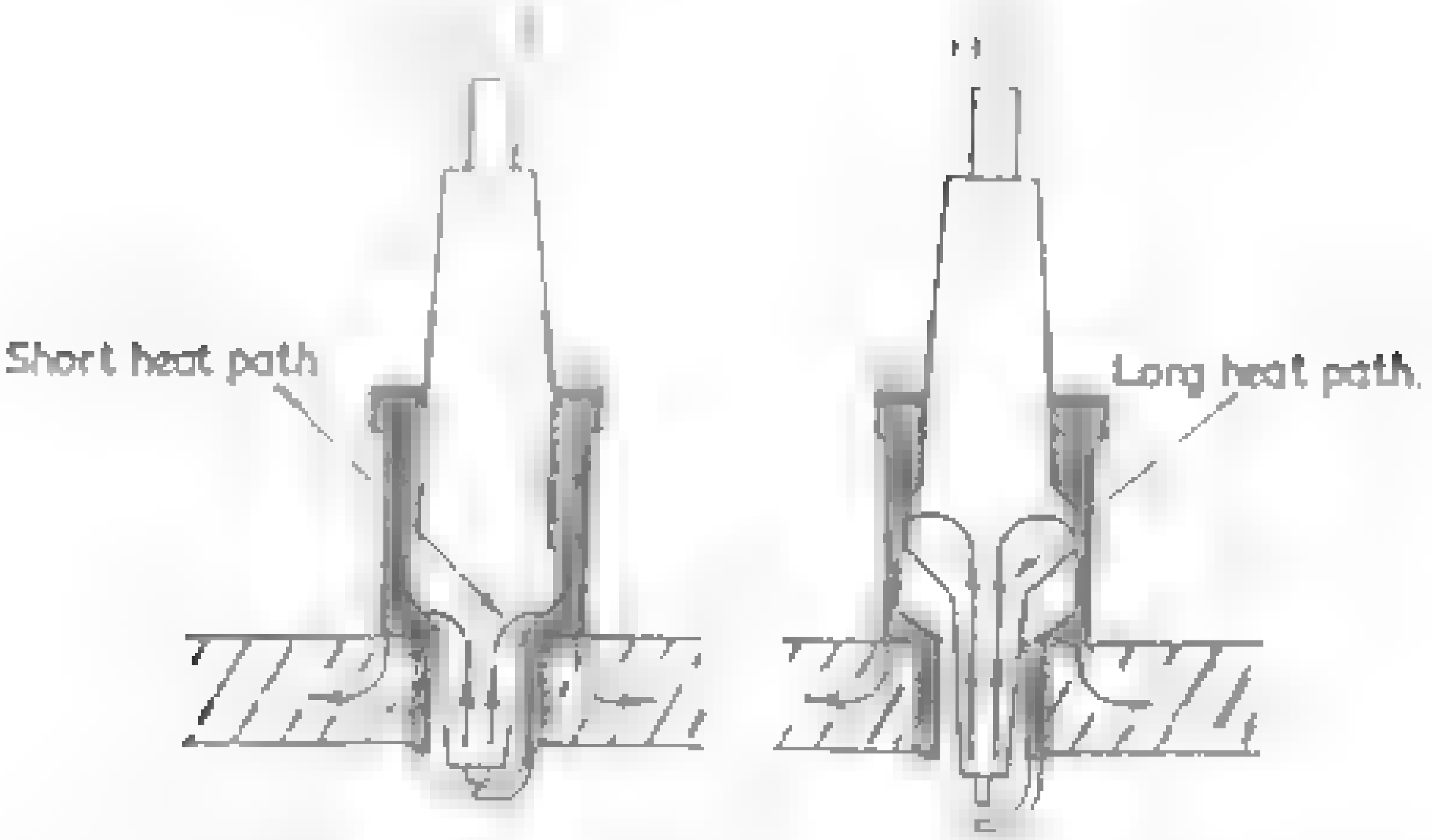


Figure 7.6 The heat rating of a spark plug is dependent on the length of the insulator nose and the electrode material composition

combustion chamber or cylinder head temperatures, ie in a relatively low horsepower motor. A cold plug, on the other hand, transfers heat rapidly from the firing end and is used to avoid overheating where temperatures are high, as in a racing engine.

The length of the insulator nose and the electrode alloy composition are the primary factors in establishing the heat rating of a particular plug. Hot plugs have long insulator noses, and therefore a long heat transfer path, while cold plugs have shorter nose lengths to transfer heat more rapidly from the insulator tip to the water jacket via the metal spark plug body. (Figure 7.6)

Motors in sports tune will probably require a standard plug or one not more than a couple of steps colder than standard. Semi-race engines cause us more problems as two plug types may be needed to cope with city driving and high-speed running. A plug two or three grades colder than standard should be tried for everyday street use and short full throttle bursts; it must also be able to resist low-speed fouling and pre-ignition at a constant 80mph. The second plug type would be used under competition conditions or sustained high-speed driving.

Race engines also require two plug heat ranges, one for engine warm-up and one for racing. If there is sufficient space in the combustion chamber, projected nose or regular gap plugs are best used for warm-up, otherwise use the hottest retracted-gap racing plug available. Whenever warm-up plugs are fitted, tag the tacho or steering wheel as the engine could be destroyed if, in the heat of the moment, your memory lets you down and you race with hot plugs fitted.

After you have decided on a plug of the correct heat range, you must test to ensure that your choice is the right one. Providing that the engine is in good condition and the carburettor is correctly tuned, reading the nose of the plug will indicate if it has the correct heat range. To avoid engine damage, it is always advisable to begin testing with plugs that are too cold, or else test the machine at moderate load and speed, then check the plugs before you engage in any full-power running.

For the plug reading to be accurate it will be necessary to run the engine at full throttle and maximum speed on the track (or road), then cut the engine dead. If you

allow the engine to keep running as you bring the vehicle to a stop, the plug reading will be useless.

As I pointed out in an earlier chapter, we are not just interested in the colour of the insulator nose. Some fuels will not colour a plug, and it takes miles for many types to colour the insulator. Therefore all of the plug firing end exposed to the combustion flame must be examined and read. The signs to look for are indicated in Table 7.6.

Of course, for those involved in road racing or long and short course speedway racing, the spark plug heat range must be tailored to each race circuit. Tracks with long straights or high-speed banked turns usually require a colder plug than a circuit with short straights and many esses and hairpin corners.

When the fuel blend is changed, the plug heat range will have to be changed accordingly. In unsupercharged applications, the plug will probably have to be one grade colder for each 10–15% of nitro added.

Table 7.6 Spark plug readings for correct heat range and other conditions

Spark plug condition	Indications
Normal – correct heat range.	Insulator nose white or very light tan to rust brown. Little or no cement boil where centre electrode protrudes through insulator nose. Electrodes not discoloured or eroded.
Too cold – use hotter plug.	Insulator nose dark grey or black. Steel plug shell end covered with dry, black soot deposit that will rub off easily.
Too hot – use colder plug.	Insulator nose chalky white or may have satin sheen. Excessive cement boil where centre electrode protrudes through insulator nose. Cement may be milk white or meringue-like. Centre electrode may 'blue' and be rounded off at edges. Earth electrode may be badly eroded or have molten appearance.
Pre-ignition – use colder plug and remove combustion chamber deposits.	Insulator nose blistered. Centre electrode and side electrode burned or melted away.
Detonation – retard ignition and richen mixture.	Fractured insulator nose in sustained or extreme cases. Insulator nose covered in tiny pepper specks or even tiny beads of aluminium leaving the piston. Excessive cement boil where centre electrode protrudes through insulator nose. Specks on plug shell end.
Insulator glazing – replace with plugs of same heat range. If condition reoccurs fit plugs one grade colder.	Shiny yellow, green or tan deposit on insulator nose particularly close to centre electrode.
Ash fouled – clean or replace with plugs of the same heat range.	Thick yellow, white or light brown deposit on insulator, centre and side electrode.

Once you have determined the correct plug heat range, do not swap over to another brand with an 'equivalent' heat range. Heat range conversion charts should be used as a guide only when you swap from one plug brand to another, as different plug manufacturers use different methods of determining the heat range of their plugs. If you cross-referenced the conversion charts from all the plug manufacturers, you would find that they disagree with each other, due to different test procedures.

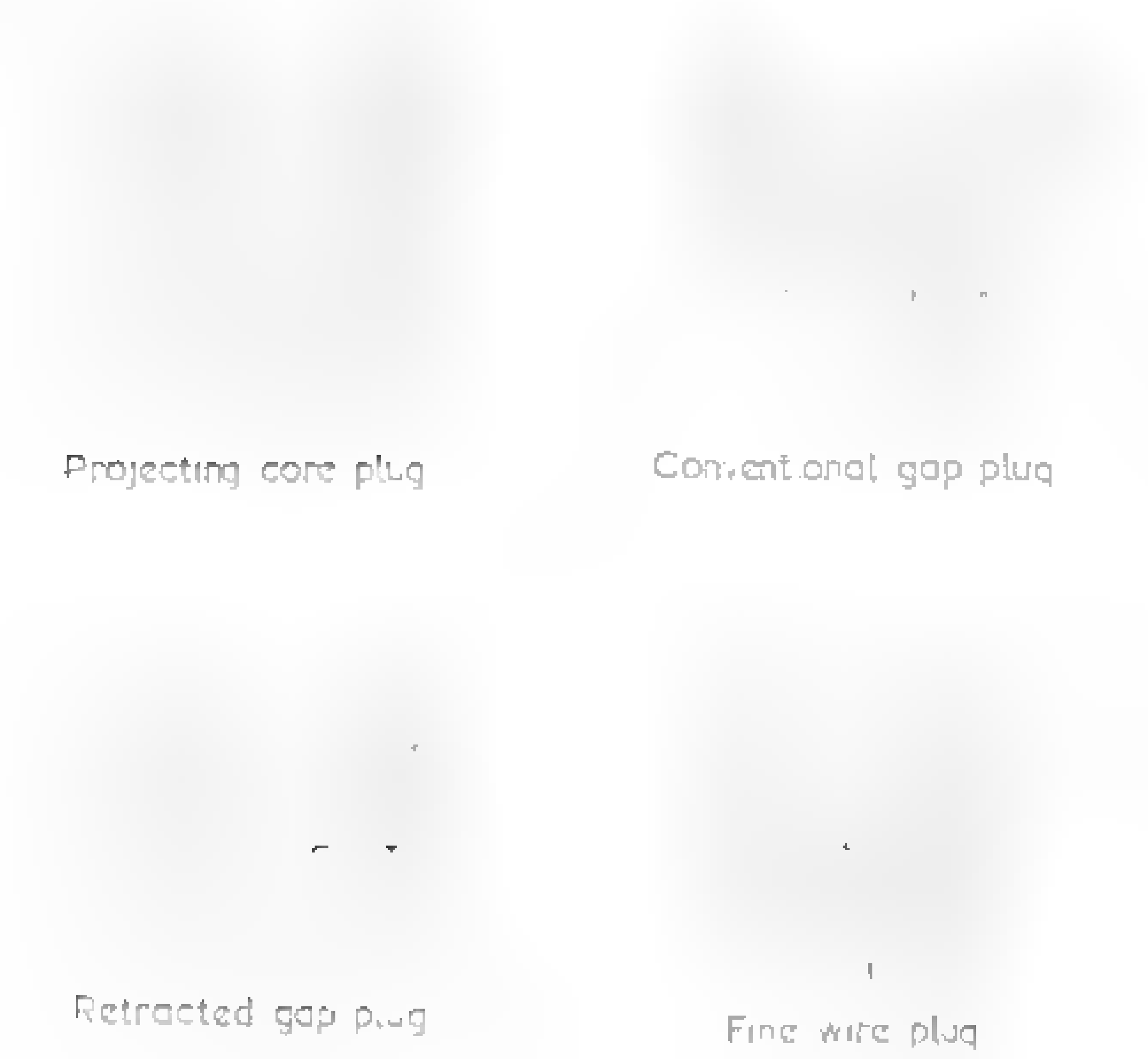
Even spark plugs with the same number from a particular manufacturer can have a wide heat range tolerance. This is why some mechanics are very particular about the brand of plug that they use in their engines. They claim that one particular brand gives a more consistent heat range, so they do not have to test dozens of plugs with the same number to find a set that has the same heat range.

SPARK PLUG GAP STYLE AND ELECTRODE MATERIALS

As well as the heat range, the gap style of the plug must be considered to obtain the best [illegible], and in some instances to avoid mechanical engine damage (Figure 7.7).

The best plug to use in most engines, where there is sufficient physical space between the tip of the plug and the valves and piston crown, is the projecting nose, or projecting core, type with copper implants in both electrodes, to increase heat transfer

Figure 7.7 Spark plug gap styles



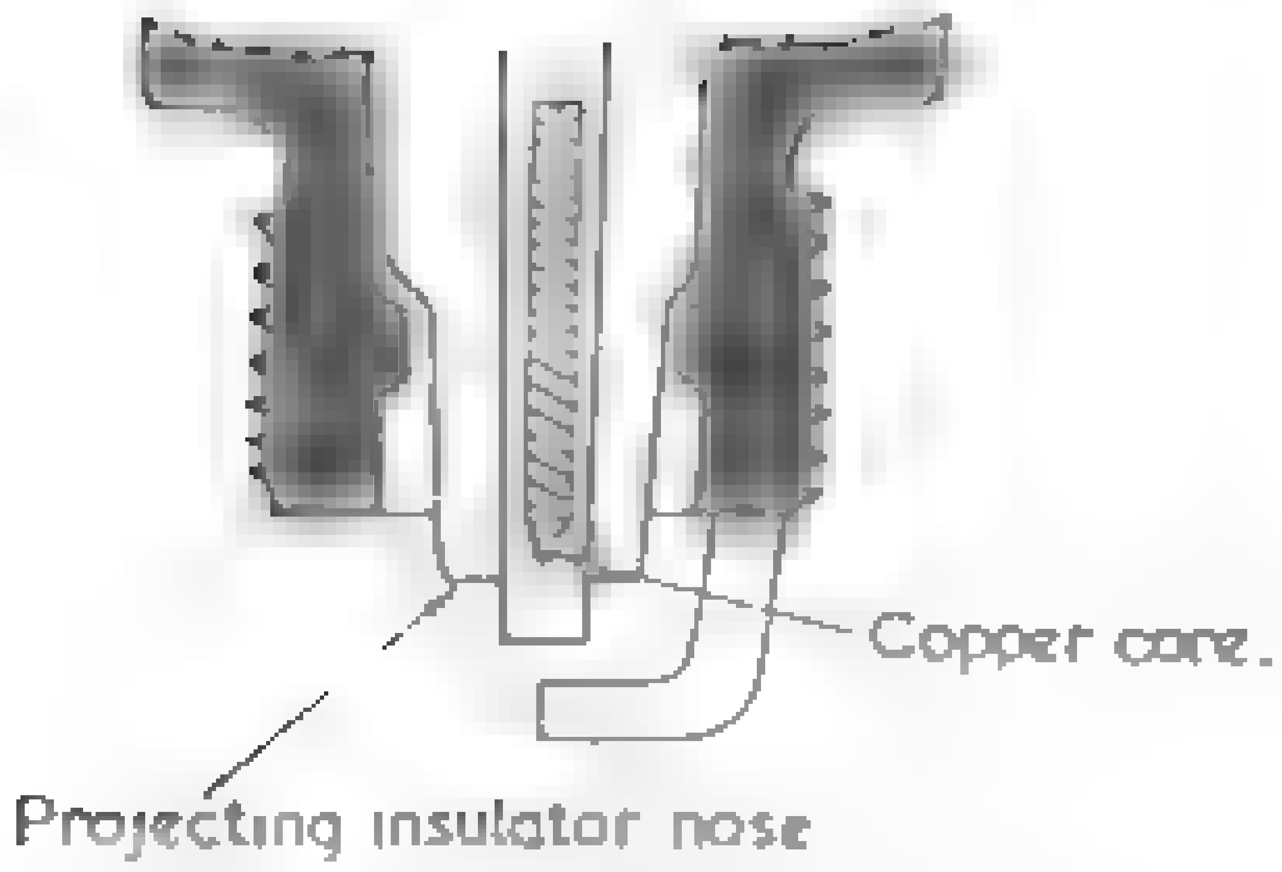


Figure 7.8 Copper implants in the spark plug electrodes increase heat transfer rates giving both projecting nose and regular gap designs a wider heat range

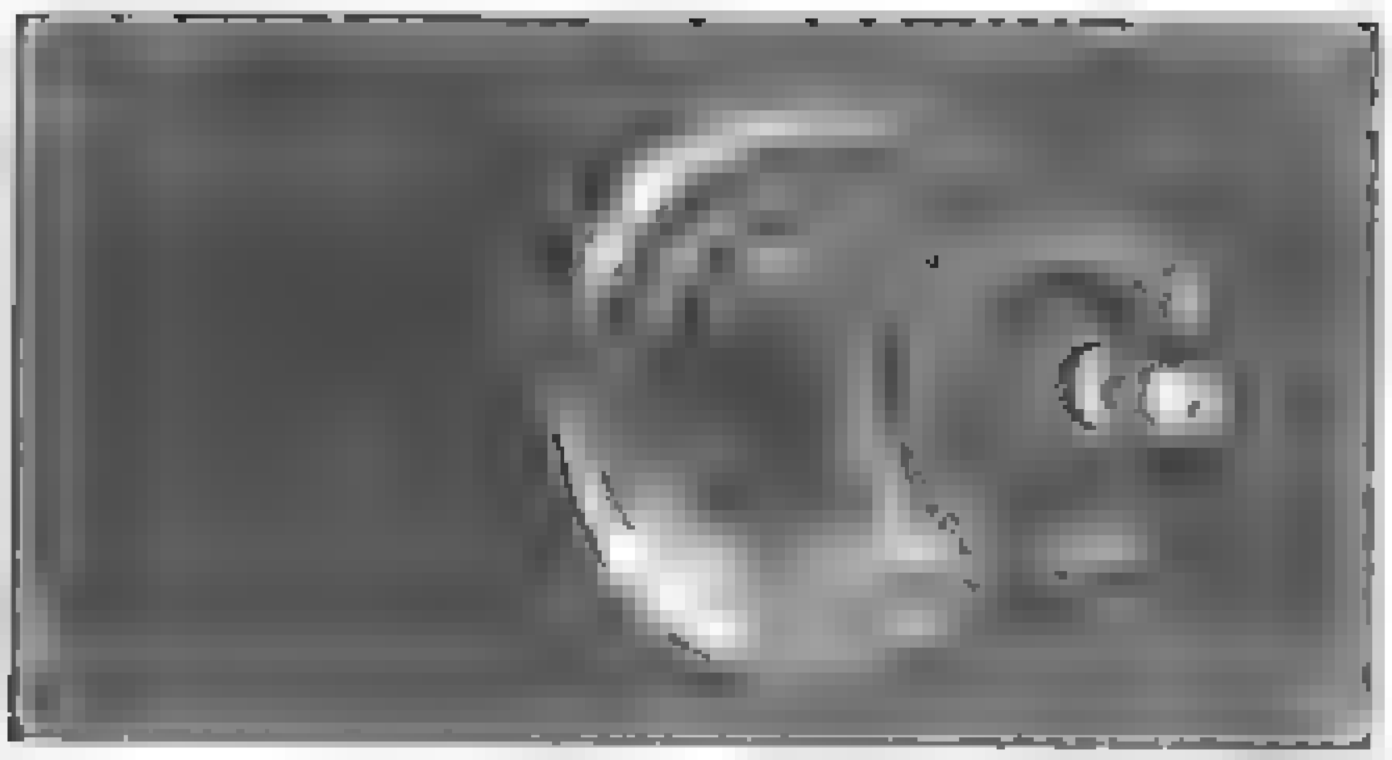
rates (Figure 7.8). It has a very wide heat range to resist both fouling and pre-ignition. At high speeds the long insulator nose is cooled by the incoming fuel charge to increase its cold heat range, while at low speeds the long tip runs hotter, to prevent fouling. Projecting nose plugs are not recommended for highly supercharged engines or those using more than 20% nitro fuel.

If you change from a regular-gap or retracted-gap plug to a projecting nose type it may be necessary to retard the advance slightly. The projecting nose starts the ignition flame burning physically deeper in the combustion chamber and this reduces the length of flame travel within the chamber. Often, projecting nose plugs will raise the power output of the engine right through the power range because of improved combustion.

The conventional-gap plug is next preferred after the projecting nose type, and can be used in engines not able physically to accommodate projecting nose plugs. This style has a wider heat range than a retracted gap plug and provides superior ignition flame propagation.

The retracted-gap racing plug is necessary in highly modified, supercharged or high-percentage nitro-fuelled racing engines. It may also be used in racing engines where there is insufficient clearance between the spark plug and valves or piston to use either a projecting nose or regular gap design. A plug of this type should be used

Carefully check the tips of all spark plugs for the telltale signs of detonation. Use at least a 4X lighted magnifier



Filing the earth electrode back to the middle of the centre electrode and tapering the corners has two effects. Reducing the length of electrode exposed to combustion heat makes the plug run cooler by around half a grade. More significantly, because there is less earth electrode in the way, the spark initiated flame kernel moves out through the entire fuel air charge more uniformly, thus giving faster more complete combustion

only when absolutely necessary. It has very little resistance to fouling and generates a poor ignition flame front.

The fine wire plug (eg Champion Gold Palladium) has some use in semi- and full race engines. It was originally developed for racing two-stroke engines but can be used in four-stroke applications requiring a wide heat range (similar to projecting nose plugs) in the colder grades. These plugs are expensive, due to the use of a fine semi precious metal electrode, but they perform very well in high-performance and race engines requiring a cold plug relatively resistant to low-speed fouling. They are not recommended for highly supercharged or high-percentage nitro-burning engines. Also as many fine wire plugs utilise a platinum electrode care must be exercised when burning methanol. Platinum acts as a catalyst in the presence of methanol and may cause pre-ignition. Some racers have been blissfully unaware of this and not had any problems, but others have not been so fortunate.

SPARK PLUG REACH

Whenever a cylinder head with larger valves, modified combustion chamber, or relocated spark plug is used, or when pistons with a non-standard crown are [illegible] always check that the spark plug is of the correct reach and gap style to avoid mechanical damage to the engine.

With the head removed, check that the plug reach (ie the length from the plug seat to the end of the thread) is neither too long nor too short. A plug that does not extend the full threaded length of the plug boss in the head will reduce performance [illegible] masking the ignition flame and invite hot spots from carbon build-up in the unused thread. A plug that is too long (ie has exposed threads in the combustion chamber) can cause damage and stripping of the plug boss threads in the head as carbon build-up on the exposed plug threads makes plug removal very difficult. Also the exposed threads can become a hot spot to initiate pre-ignition. In some instances a change to a plug with a different reach may be in order, but in most cases the use of a single sol[illegible] [illegible]sket on each plug will ensure the proper depth fit. The range of Champi[illegible] gaskets is listed in Table 7.7.

Table 7.7 Champion spark plug gaskets

Plug diameter (mm)	Gasket thickness (in)	Part No
10	0.045	Y-674
	0.055/.045	Y-678
12	0.057	P-674
	0.070/.052	P-678
	0.095	P-677
14	0.057	N-675
	0.135	N-677
	0.070/.052	N-678
	0.080	N-673X1
18	0.080	A-675
	0.065/.050	A-676
	0.075/.055	A-678

With the head fitted, check that sufficient clearance exists between the plug and the piston. To allow for rod stretch the normal plug gap should be increased by 0.025in. Turn the engine over by hand, then remove the plugs and inspect them for closed gaps. When carrying out this check remember that a hot projecting nose plug extends further into the combustion chamber than a colder one, so be sure to make the check using the hottest plug that will be used in the engine.

SPARK PLUG GAP

The width of the spark plug gap for best performance depends primarily on the compression pressure of the fuel/air mixture, the engine speed and the coil saturation time. Increasing the first two factors and decreasing the latter calls for a decrease in plug gap. It is safe to say that all modified engines using factory ignition require a plug gap narrower than stock.

Manufacturers stipulate a relatively wide gap as this improves the engine idle and low speed performance. The ignition spark from a wide gap is larger than that from a narrower gap, so a larger initial combustion flame is generated, assisting low-rpm running. However, at high engine speeds and compression pressures the coil is not able to supply electrical energy of sufficient intensity to jump the spark gap, so the gap must be reduced to avoid a high speed misfire, or the possible breakdown of the high-voltage system insulation.

From experience I would recommend that no high-performance engine using a conventional high-energy electronic ignition should use a plug gap exceeding 0.035–0.040in. Semi- and full race engines may need a gap as small as 0.028–0.032in with conventional plugs, and down to 0.025in when using retracted gap racing plugs. Race engines running methanol will require gaps 0.005–0.010in smaller than these figures because it is much more difficult to get methanol burning initially, then keep it burning for complete combustion.

Engines with a capacitor discharge ignition can pick up some power with wider plug gaps as this type of ignition is relatively insensitive to high engine rpm. In this instance the plug gap should be set to the ignition manufacturer's specifications for

short-distance events and ordinary road use. However, those involved in long-distance racing and rallying or constant high-speed road driving should not use a gap exc 0.045in, as any weakness developed in the CD unit during a race could lead to total ignition failure. A closer plug gap reduces the load on the 'spark box', minimising the chance of any weakness developing, and if some trouble does arise, the system may have just enough reserve to finish the event.

COIL POLARITY

Always ensure that your coil polarity is correct before you try experimenting to determine what plug gap gives the best performance. The equivalent of 40% coil energy is lost when the polarity is reversed.

The spark should always jump from the centre electrode of the plug to the side electrode as this considerably decreases the voltage required for ignition. Due to the centre electrode being at a much higher temperature than the side electrode, less voltage is needed to produce a spark, as electrons will leave the hotter surface at a lower voltage. This is why it can be difficult to get a cold motor started, because the centre electrode is cold, a much higher voltage is required to produce a spark.

The coil polarity can be checked by looking at the coil's low-voltage connections (electricity supply wire from ignition switch to positive; module/amplifier wire to negative), or by using an oscilloscope. The oscilloscope's trace pattern will be upside down if the polarity is not correct. A dished or eroded spark plug side electrode also indicates wrong polarity; the dish is caused by metal leaving the electrode each time a spark jumps across to the centre electrode.

SPARK PLUG MAINTENANCE

The life of a spark plug is not as short as many would suppose. Some drag racers believe that a new set of plugs is required for each quarter-mile trip, but cars involved in 500-mile speedway races and 12- and 24-hour road events go the full distance without a plug

A centre electrode eroded like this indicates correct coil polarity and shows that a good strong spark is being produced. However, with lead deposits on the insulator nose and the gap widened like this, the plug should be replaced.

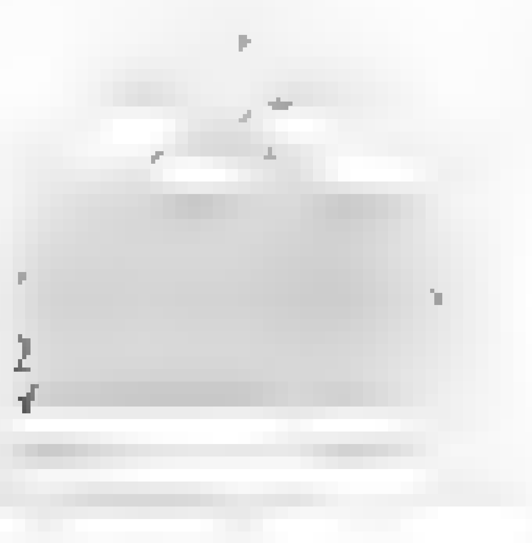


Figure 7.9 Spark plug electrode filing

cannot be filed, and fine wire electrode plugs should not be filed

the threads

Personally, I prefer to leave plugs uncleaned. If they are fouled by fuel or oil I clean them with a toothbrush and ether, or some other non-oily solvent. If the insulation is breaking down due to leaded fuel deposits, I throw the plug away.

SPARK PLUG LEADS

The spark plug leads provide the electrical connection between the distributor cap and spark plugs. Most stock road cars use radio suppression cables with a powdered carbon impregnated rayon cord to conduct this high-voltage current, which will

not spark. Eventually one or more of the plugs will not fire at all. Therefore carbon suppression cables should be discarded and replaced by high-quality induction type suppression leads. These leads have a metal core to conduct full electrical energy to

the plugs, and they have a metal induction spiral wound within the cable to provide both noise and induction suppression. Remember that inadequate suppression could seriously upset the proper functioning of the ECU, and other electronic or radio-type devices, so do not use ordinary solid copper core or stainless steel core cable.

The other good reason for using induction spiral leads is to minimise the [illegible] energy ignitions. Basically, due to magnetic induction, whenever high-voltage wires are grouped closely together, or run in parallel for any distance together, one with current flowing through it can induce a corresponding voltage in a neighbouring wire. In fact [illegible] going on between plug wires! Induction crossfire in its mildest form causes the engine to run roughly, but in situations of high engine load or high rpm such as are typical under competition conditions, it can result in serious engine damage.

To help you better understand the seriousness of the problem, let us take a look at a typical V8, as these engines most frequently experience a crossfire problem. A crossfire that will destroy cylinder walls, pistons, bearings, cranks, heads or head gaskets is most likely to occur between consecutive firing cylinders when these are [illegible] 125° BTDC, and it is almost full with fuel air mixture. At this stage No 7 piston is moving up on the compression stroke and, depending on the cam profile, the inlet valve is closed or at least about to close. It is in this setting that No 5 plug wire has a [illegible] flow in the No 7 plug wire, that plug also fires, but as it is 90° early violent combustion occurs. Depending on how often this happens, and under what sort of [illegible] a wrecked block or crankshaft.

The way to avoid this sort of problem is to use induction spiral suppression [illegible] once. Crossing the problem wires once inverts the magnetic field in the second wire [illegible] also open the way for misfiring.

For the plugs to fire the plug wires have to prevent the coil's high-voltage current [illegible] on running race engines with big plug gaps, plug wires need all the help you can give them. [illegible] run into a misfire problem. With most race engines pushing out about [illegible] cylinder, that means you lose 50–80hp each time the spark plug fails to light.

The average race engine needs at least 20,000 volts, and anything up to 35,000 volts if the plug gaps are very wide, to get a spark across the spark plug. When you compare the amount of insulation that you have available to prevent all that electrical pressure flashing across to earth with what the power companies use on the overhead

power lines, you begin to appreciate what you are up against. To work in this sort of environment you must use only top quality silicone cable and boots. I recommend 8mm silicone wire for the majority of road applications and race engines that will not see voltages much over 20kV. This means plug gaps down around 0.028in and an 11:1 [illegible] and higher we have to be careful to choose only the best grades of ignition wire. Some people think that 9mm or even 10mm wire is the solution. However, 8mm Moroso spiral core wire has a dielectric insulation strength of well over 50kV, which is superior to anything else I know of, and it combines the highest levels of electro- [illegible] energy at the spark plug.

DISTRIBUTOR CAP AND ROTOR BUTTON

Besides the plug leads, other potential high-voltage leakage or flashover areas are the distributor cap, rotor button and coil tower. These parts must be free of accumulated dust, [illegible] Therefore a periodic check must be made to ensure that none of these conditions exist.

Unknowingly, some tuners bring trouble upon themselves by removing carbon from the distributor cap terminals and the rotor button contact. Actually, carbon in these areas assists electrical conduction. The real problem is that quite often the knife or screwdriver being used to scrape the carbon from the terminals slips and scratches the glaze inside the cap. This drastically reduces the dielectric strength and can lead to a carbon track forming during wet conditions. Often a piece of abrasive paper is used to polish the rotor button contact. This cuts straight through the insulating glaze and results in voltage leaking to earth.

When high-voltage ignition systems are used, particular attention must be paid to keeping all the insulating mediums clean, otherwise less voltage will be available to fire the plugs than when the standard ignition system was fitted. Actually, when a [illegible] alkyd material, should be used, being resistant to carbon tracking even when cracked.

As mentioned earlier, another thing we have to worry about with high-energy ignition is the problem of crossfiring within the distributor cap. This is usually not a worry with four- and five-cylinder engines, but it begins to surface with six-cylinder engines and becomes a real difficulty once we get to eight or more cylinders because the high-voltage posts are just so close to one another. The first move is to swap to a larger diameter cap containing well-designed anti-flashover ribs; this will put more space – a larger air gap – between the posts.

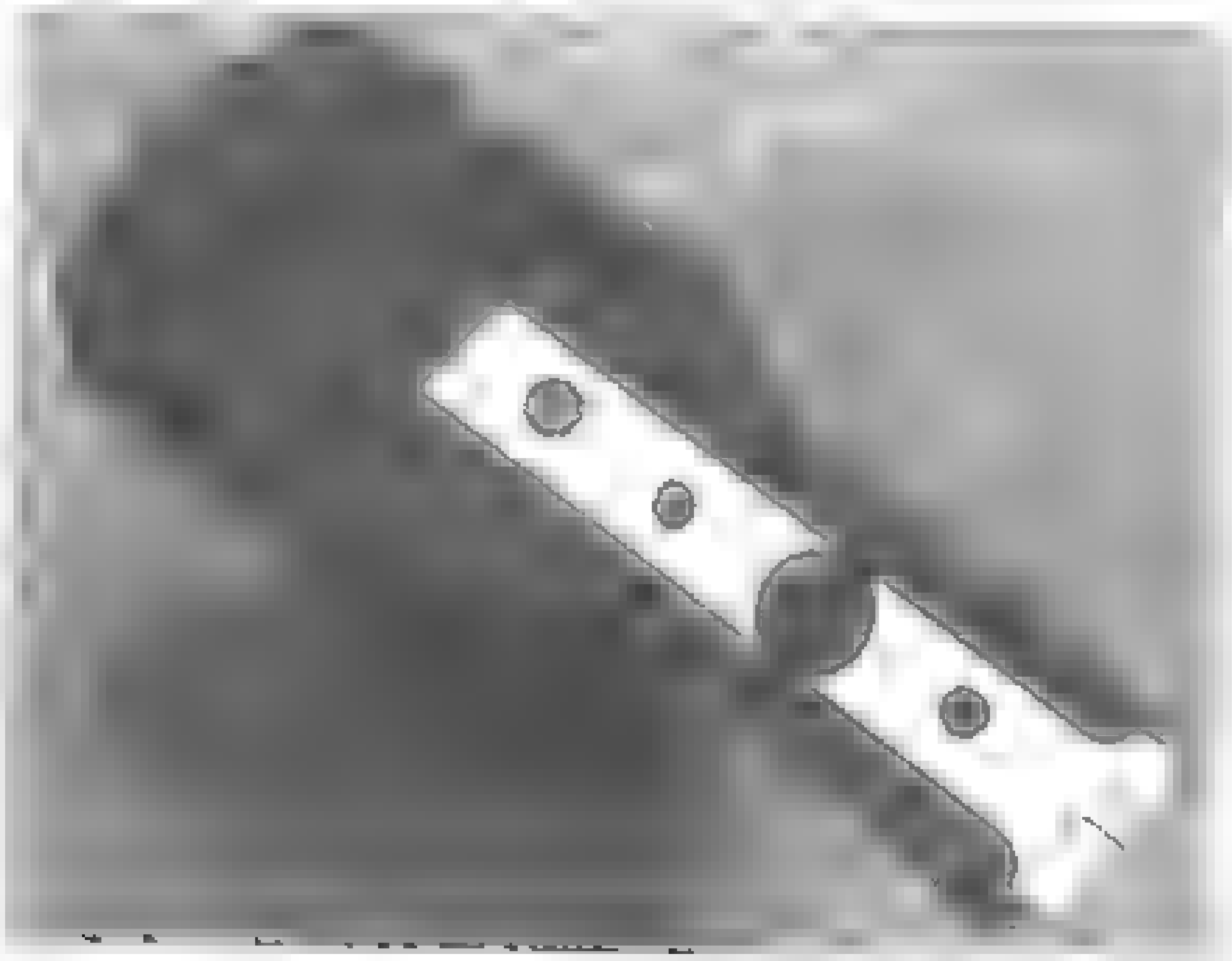
However, the location of the rotor button when the ignition fires, and the length of the rotor button's brass tip, also have a bearing on internal flashover. We tend to think [illegible] long brass tip on the rotor button to help place it somewhere close to the distributor cap post when the ignition fires. This is not a problem when ignition voltage is low, but as we begin to increase electrical pressure in the system to shoot a spark across a plug gap in a high pressure environment in the combustion chamber, the [illegible]

Now the easiest path to earth – and remember that electricity always takes the easiest path – may not be across to the distributor post to which the rotor button is closest and down the plug wire. Instead, that wide brass tip on the rotor button may be closer – not physically, but from the aspect that the electrical resistance is less – to a distributor cap post either directly in front of, or else directly behind the cylinder that is supposed to be getting ignition energy. When this situation exists the electricity jumps from the tip of the rotor across to a post that may be half an inch or more away, then travels down the plug wire to fire a plug in that empty cylinder. Because it is not that cylinder's turn to fire, there is less pressure in it, so it is easier for the electricity to get to earth via that route, even though it has to jump a much bigger gap to get across to the distributor terminal for that cylinder.

There are a number of ways to minimise this sort of flashover. If it is possible the easiest way is to cut back on the length of the rotor button's brass tip. To check this remove the distributor cap and turn the engine over to 10° before TDC, then accurately scribe a line on the brass tip to correspond with the centre of the plug wire post. Move the crank back to 35° BTDC and again scribe the rotor button to correspond with the distributor cap post. The brass tip extending beyond the boundary of those two scribed lines can now be cut off, thus widening the air gap to the adjacent plug wire terminal in the distributor cap.

When doing this check you may be surprised to find that the rotor button is way off lining up with the plug wire post. Here there are two options: relocate the rotor or relocate the distributor cap. If you choose to relocate the rotor, have a new locating slot cut into the distributor shaft. Some enthusiasts knock out the locating tab of the rotor, then either epoxy or pin the rotor to the distributor shaft, but this is not always successful.

The brass contact on this rotor button has been shortened at both ends to prevent crossfiring within the distributor cap.



Another way to reduce flashover within the distributor cap is to vent it of ionised air What is needed is a 10mm hole between each plug wire terminal around the circumference of the distributor cap, and at about the same height in the cap as the rotor

This brings us to the final solution when we are forced to run a distributor-type [illegible] choose to take this path of less resistance and fire the plug as it is supposed to, rather than crossfire to another cylinder within the distributor cap

PRIMARY VOLTAGE INFLUENCES IGNITION PERFORMANCE

Before we finish with the subject of ignition systems, we need to give some [illegible] we should look at the idea that pushing up the primary-side voltage will increase the energy available to fire the plugs

The electrical energy consumed by high intensity electronic or capacitor discharge ignitions is not insignificant. Added to this, racers have over the years added more and more electricity-consuming devices to their race cars. When electronic fuel injection, electric fuel pumps, electric cooling fans, etc, became a standard part of the modern race car, loads on the electrical system increased dramatically by comparison with what they were previously. Because some racers have not given proper attention to this aspect of race car engineering, they find that their cars are down on power at best, or they may experience reliability problems, with the car stopping out on the track for apparently no reason

For example, CD ignition can have a current draw two or three times higher than a transistorised HEI system – up around 8–10 amps However, while the average power draw of an HEI system is relatively low, the instantaneous amperage draw, when the module first switches the primary circuit to charge the coil, is very high It is for this reason that a transistor HEI must be wired directly to the ignition switch using heavy 10 gauge wire, and nothing else should be connected in that circuit. Now, when the current begins to surge into the coil, that high instantaneous power drain can, if the [illegible] for an instant However, it can occasionally cause the ECU to shut down completely – then the engine stops Incidentally, I have seen the same sort of thing happen when the electric cooling fan cuts in and sends a surge through the electronics

Also, the idea that increasing the system voltage from, say, 12 volts to 16 volts, to achieve a bigger output from CD ignition, is a fable. All the CD systems I know give a constant output regardless of whether the feed voltage is 10 volts or 16 volts. If you see increased ignition output with CD systems when the supply voltage is increased, you can be certain that you have a problem in the primary feed circuit Either the cable is too small or too long, or you have 'hot' joints with cables poorly crimped to connectors, or connectors not making perfect contact and cutting supply voltage at the amplifier to less than 10 volts

Transistor HEI systems, however, are very different in this respect. Any reduction in supply voltage will cause a corresponding reduction in coil output

Conversely, any increase in system supply voltage will provide an increase in secondary coil energy. As a minimum we should be expecting the ignition supply voltage at the coil positive (+) terminal to be the same as the battery voltage. Thus if the battery measures out at 12.6 volts, we should see the same voltage at the positive coil terminal with the engine running and all of the car's electrical equipment turned on. If the voltage is less than the battery voltage, you are losing out on maximum spark energy, so increase the size of the ignition circuit wiring and ensure that nothing else except the ignition is connected in this circuit. Note that you must take care when carrying out this test as there could be 350 volts at the coil negative (–) terminal.

To give you some idea of how low voltage affects transistor HEI ignition spark intensity, and consequently engine hp, note what happened during the testing of a 350 Chev. The engine had been rebuilt with new exotic CNC ported hand finished cylinder heads which it was hoped would lift power output to around 650hp. However, after being run-in the first few pulls were showing nothing better than 570hp. With some fiddling during an entire day this finally rose to a disappointing 582hp, 40hp less than the previous pair of heads with minimal CNC porting.

The following day started out even worse with the engine sounding sick, and the printout proved it. After a lot of checking the problem was isolated to a faulty new race battery which was almost 4 volts down on the rated output voltage. With a new up to spec battery connected the engine note immediately changed to a healthy crisp sound. After a few pulls to optimise the fuel and spark the engine made 646hp. Based on these figures you can see the low battery voltage – and the available peak current flow would have been way down too – was costing something like 10% hp.

Many transistor HEI systems will accept higher supply voltages and thus provide more secondary spark energy to fire the plugs at higher engine speeds, or else fire bigger plug gaps or exotic fuel. For example, a V8 system that works at 7,500rpm on a six-cell battery (up to 12.6 volts) will work up to about 8,200rpm with a seven-cell battery (up to 14.7 volts), and up around 8,800rpm when connected to an eight-cell battery (up to 16.8 volts). Remember to check with the ignition manufacturer before you increase the input voltage, otherwise you may overload some part of the system and 'blow' the electronics.

When purchasing a higher-voltage battery obtain one with extra terminals that allow you to tap the correct voltage supply for various electrical items on the car. Thus while the ignition may work adequately at 16.8 volts, or even up to 18 volts, with the exception of the starter motor nothing else will survive for any time at this level. The fuel injection pump also pumps more fuel and at higher pressure at higher voltage, but most become unreliable at more than 14 volts. The remainder of the electrics, however, should not be operated at more than 13 volts.

Chapter 8

The Engine Management System

T[illegible] services far beyond basic control of the fuel and ignition requirements of the engine. With today's very specialised engine management systems you can almost specify anything you want, money permitting! The optional features available include spark and fuel trim to permit ignition timing and mixture adjustments to each [illegible] control, etc

In road cars, and sometimes in competition cars too, there are some other [illegible] cooling fan control, idle air stepper motor control; closed loop Lambda control, muffler bypass control, inlet tract switching control, cam phasing control.

While it is obviously very necessary to focus on engine management features [illegible] unit because it is the only one with a feature that we feel we must have. The problem could be that that particular system demands a huge investment of time both on the dyno and on the track to get it set up, and if that input isn't given it will cause the engine to run worse than a more basic system. Another problem can be that the management system is just so specialised that there are only a handful of people about who actually understand how to properly programme the thing. If you can't find someone who knows both the engine management system and your engine you may never realise your dreams. In fact it is more probable that your dreams will quickly become a nightmare! In similar vein keep in mind that some systems are well known and understood in one country or one branch of motor sport but are unknown

elsewhere. If you are outside that circle of knowledge then it could be risky to set your focus on such a unit.

Another thing you may need to plan for is upgradeability. If you are in a very competitive league where progress is constant you will most likely want an engine management system backed by a manufacturer who will provide regular inexpensive [illegible] they are tossing out for peanuts could well be a high-end system that is more than adequate at your level of competition.

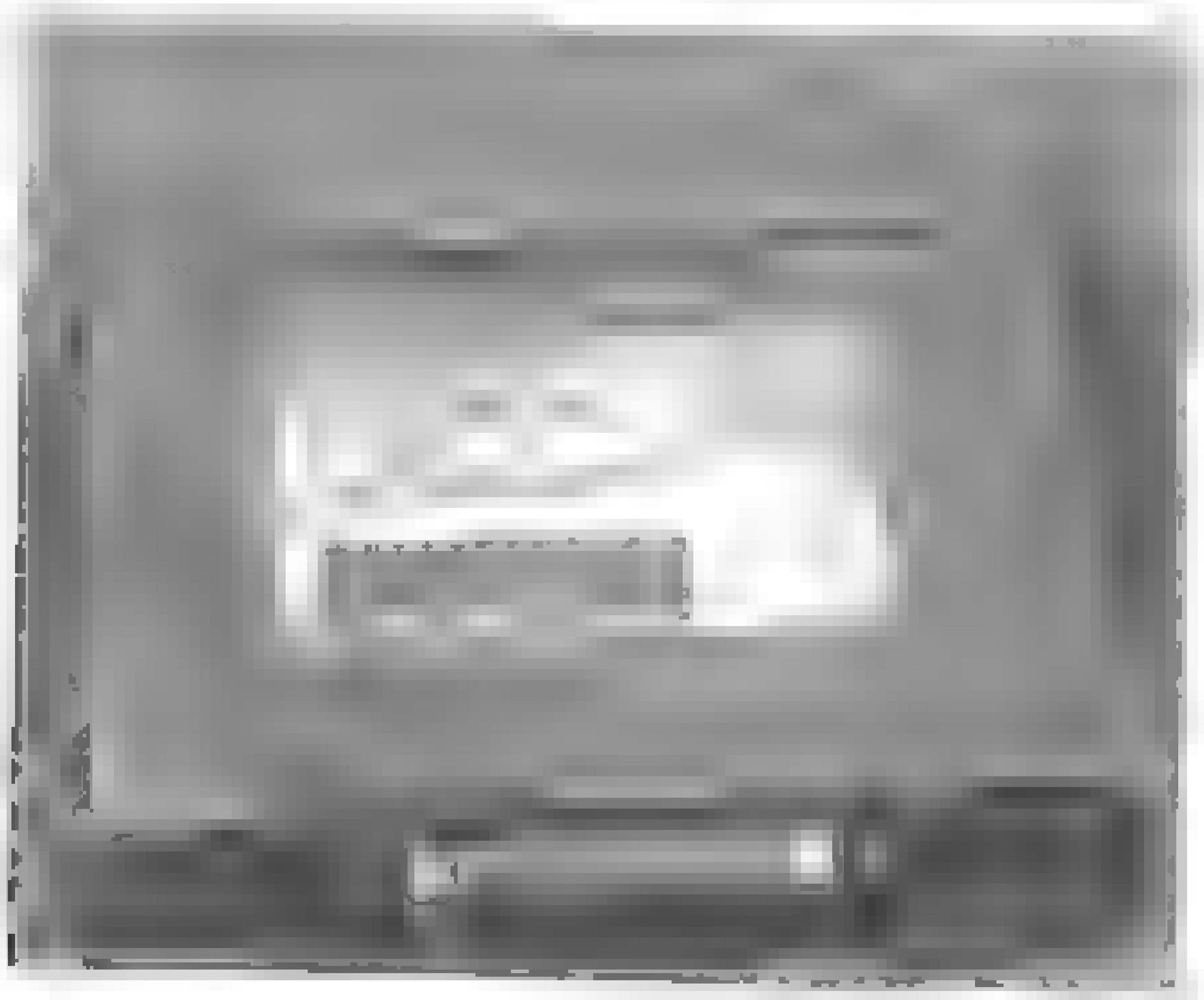
THE FACTORY COMPUTER

Very often stock engine management systems are rubbished and then replaced by inferior aftermarket [illegible] car to perform at an acceptable level. However, I question why you would want to toss an easy to programme ECU with everything you need to make street driving as pleasant as possible [illegible] things such as the idle air stepper motor or the radiator cooling fan.

Also when you decide that it is necessary to replace the factory computer don't automatically think of purchasing an aftermarket unit as you could find a number of good [illegible] ECU [illegible] car, or another model from the same manufacturer, or even from another brand of car. I know it goes against the grain for some people to do this, but it doesn't make much sense to reject a part on such grounds.

For example where I operate you can find any number of GM Delco '808' (so [illegible] same price as 5 litres of Mobil 1. Using Kalmaker software they can be reprogrammed

The Delco 808 ECU can be reprogrammed to provide excellent performance for minimal cash outlay



to work wonderfully in many applications. Most though snigger at such an idea, which is great for those in the know as there will be a ready supply of such ECUs an[d] [illegible]

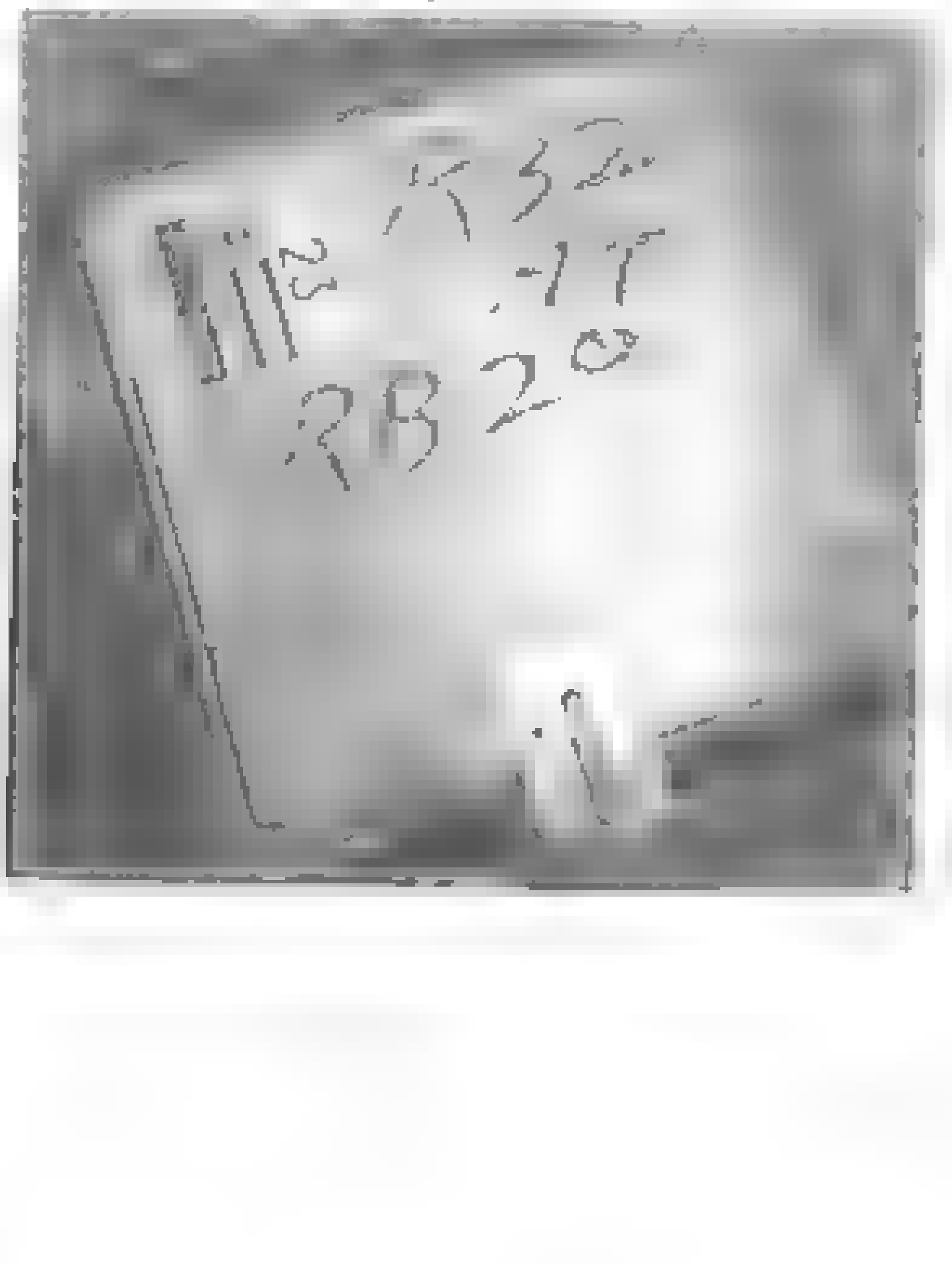
weekend competition on an exotic brew [illegible]

[illegible] upgraded very simply by fitting the later 32KB MemCal (GM speak for the EPROM chip).

Soon after this GM switched to a 64KB Delco unit which also incorporates auto trans control. They called this a powertrain control module (PCM). It also did not [illegible] mass air flow sensing, but can be switched back to MAP sensing.

With Nissans it is sometimes possible to change back to an earlier [illegible] However, for about an eighth of the price of the 'brand' Japanese computers you can obtain secondhand units out of R32s that will plug straight into the R33.

The Nissan R32 computer is programmable. This one, sourced from a 2ltr 6-cylinder model, is destined to replace the 'locked up' ECU in an R33.



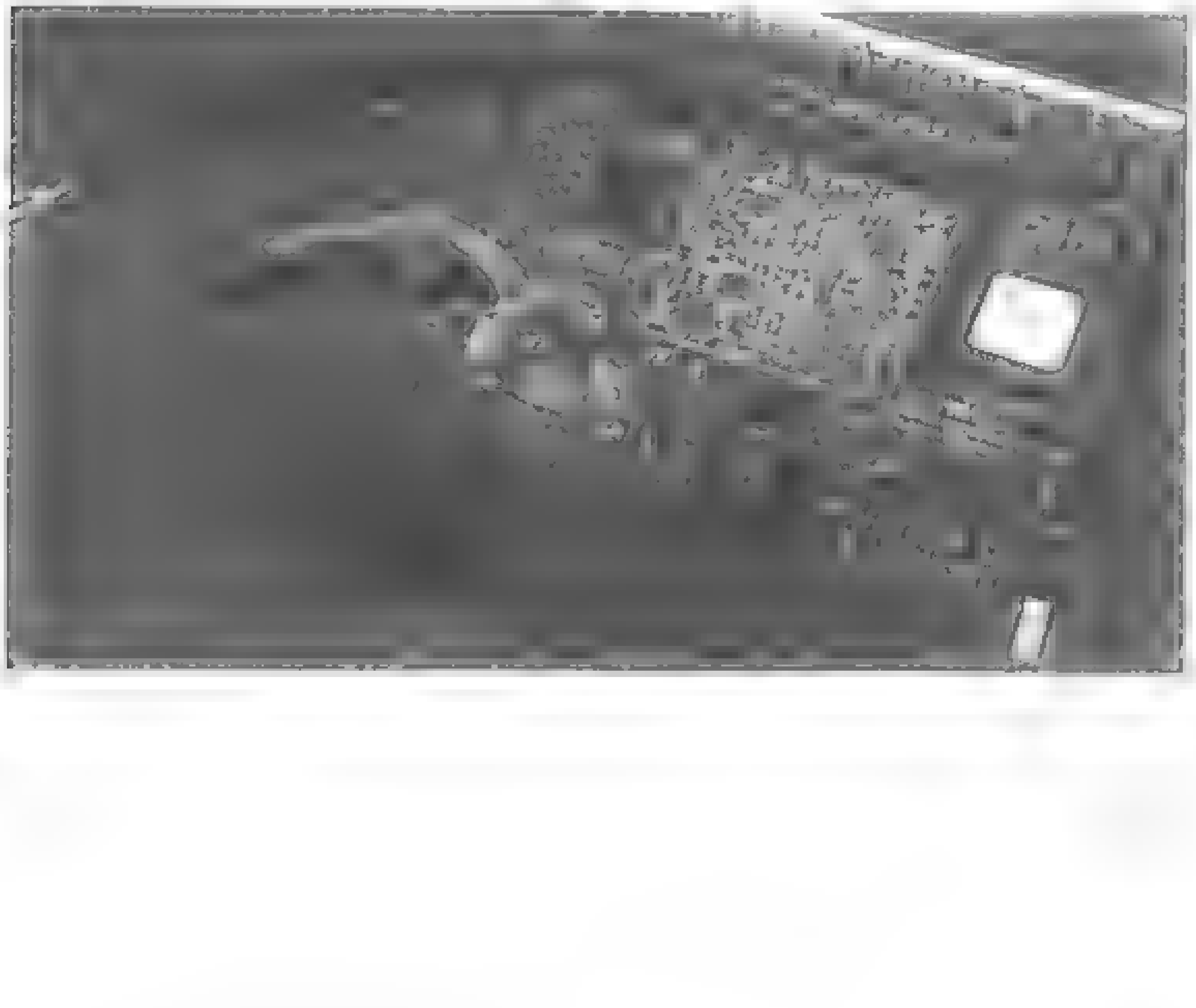
Clearly when it comes to selecting an engine management system there are a huge number of factors to consider. Unfortunately many base their decision on hardware specifications. For example, looking at the above mentioned Delco units it could be easy to conclude that the 16KB 808 was useless even as a boat anchor, and that the 128KB hardware would be the obvious choice. However hardware specifications are only a small part of th[illegible] dy mention[illegible] require a system that can be [illegible]ly programmed, but on top of that requirem[illegible] are other factors which will infl[illegible]

CHOOSING AN AFTERMARKET COMPUTER

Rather than hardware we should perhaps focus more on softwa[illegible] engine tuners don't know much about software. If they did they [illegible] industry making a lot more money. Consequently manufacturers [illegible] management equipment can't appeal to them to make purchasing decisions based on impressive software capability. Rather they have to talk to us in lan[illegible] underst[illegible] [illegible]ware sell. How many million cal[illegible]

MoTeC [illegible]

[illegible] the high spec Aut[illegible] [illegible]cts from the internal MAP s[illegible] [illegible]t manifol[illegible]



it will make per second, how it still functions in adverse environments (heat and vibration), and so forth. However as engine tuners we have to question 'So what if the [illegible] instructions issued will be useless anyway!' Really, we must never look at the hardware in isolation but rather the entire package

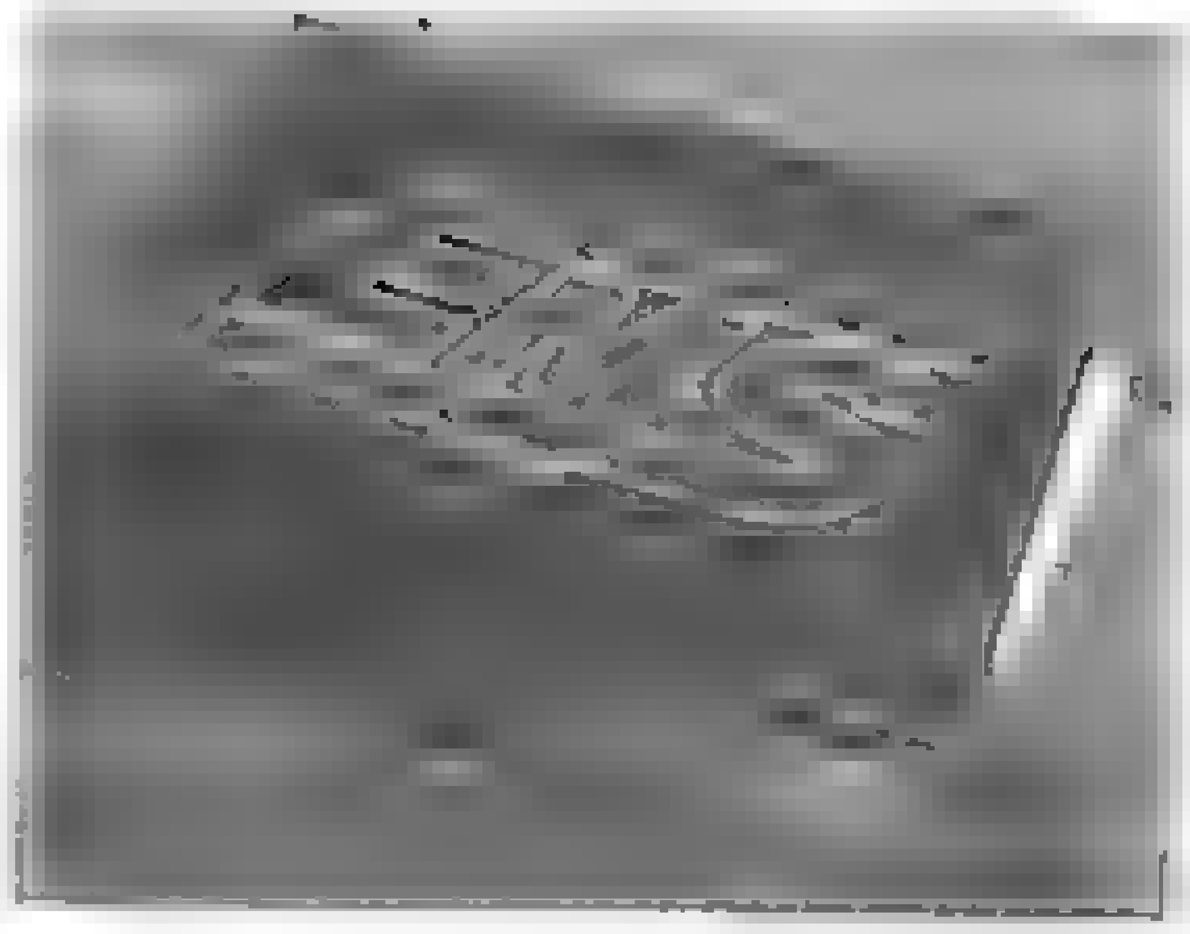
In addition we need to remember that the engine management system has to integrate with other areas of the vehicle. The Australian MoTeC system is a good example of the kind of features we would be looking for in a high-end competition management system. The MoTeC accepts a huge range of sensor inputs and will drive a diverse range of fuel injectors It has multiple injector and ignition outputs, so can [illegible] coil or multi-coil ignition systems directly from the ECU. Modern race cars rely very heavily on data acquisition and complex LCD dash displays. The MoTeC interfaces with such systems from companies like Pi Research and CIC

If we were looking at engine management purely from the aspect of engine performance, as in the numbers we see on the dyno, then the Autronic ECU will provide performance equal to, and some would say superior to, MoTeC units. The Autronic has numerous impressive features such as a full three-dimensional cold start feature which doesn't simply add a percentage increase to the injector open time but rather operates on a load versus coolant temperature map In supercharged engines the boost pressure can be controlled as a function of coolant temperature (low boost when the engine is cold) and as a function of what gear has been selected (less boost in lower gears to control wheelspin or transmission damage) Programmable outputs can switch such things as the water pump in the water to air intercooler system when the inlet air temperature reaches a certain temperature. Like numerous stock type ECUs the Autronic also has a limp home mode and it will flash out fault codes to inform you of problems without having to be connected to a laptop. However unlike the MoTeC computer there is very little flexibility in terms of sensor configuration, especially in the area of ignition pick-ups where only a few specific ignition triggers can be deciphered. Some tuners though consider this as a positive as the Autronic does not [illegible] of sensor types. Additionally for popular vehicles the Autronic can be supplied as a board which slips into the factory ECU case and connects to the stock harness, to retain a totally stock appearance

Unfortunately Richard Aubert, the man behind Autronic and previously the software expert at MoTeC, has devoted all his energy to producing and constantly refining a brilliant ECU Consequently supplied documentation isn't slick, nor easy to follow. Don't let this put you off the Autronic units though. I understand Richard is in the process of correcting this matter, but there are alternative sources. A couple of enthusiastic Autronic users, Steve Cox (www.quickfitmotorsport.com.au) and Ray Hall (www.turbofast.com.au), both with a lot of experience wiring and mapping the unit in a wide variety of applications, decided to take matters into their own hands and produce their own manuals

A mid-range engine management system, with the unlikely brand name EMS (also from Oz) has one interesting feature which adds to its user friendliness. I usually don't like, nor do I recommend units which are programmed from a hand controller. Tuning is too much a fiddly hassle for my liking Fortunately the latest EMS brand

The EMS brand electronic control unit ha... of useful features including live data while you drive. This ... mapping glitches easier to identify ... rectify. In spite of what the label seems ... not a fuel co...

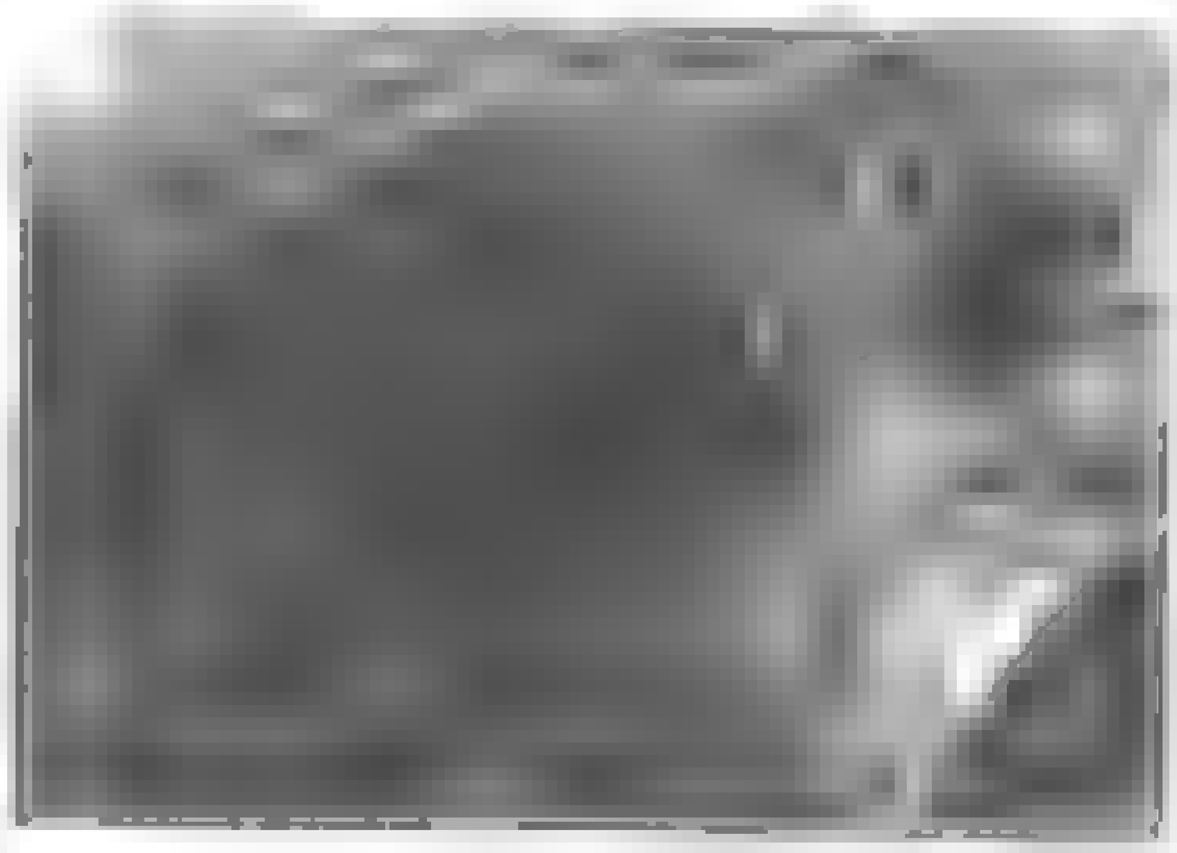


...e you the option of calibrating the ECU using either their hand controller or a laptop. I would choose the laptop route every time. However, having said that, the hand controller does have one appealing function – it can provide live-data as you drive. As such it can save you fitting extra gauges that you would otherwise need to obtain to provide you with such information as charge air temperature and the air fuel ratio. For this latter function to operate a wide-band lambda sensor must be fitted to replace the stock exhaust oxygen sensor.

Having the air fuel ratio meter function built into the hardware is unique and enables the vehicle owner to report back to the tuner any fuelling problems after the system has been initially mapped on the rolling road. Car manufacturers have a powertrain team spend months, at a cost of many millions, in an effort to get the programming correct. Obviously a tuner in a few hours on the dyno can't duplicate every conceivable engine operating condition to achieve similar results. With the EMS computer providing an air/fuel ratio readout, any tuning glitches can be fairly quickly identified as being either a fuelling problem or an ignition advance problem during normal operation on the street.

The Haltech ECU is supplied with excellent documentation to simplify installation.

Idle speed stepper motor con... is required in street vehicles. It prevents stalling by compensating for increased engine loa... whe... or when using power steering ... ight parking space



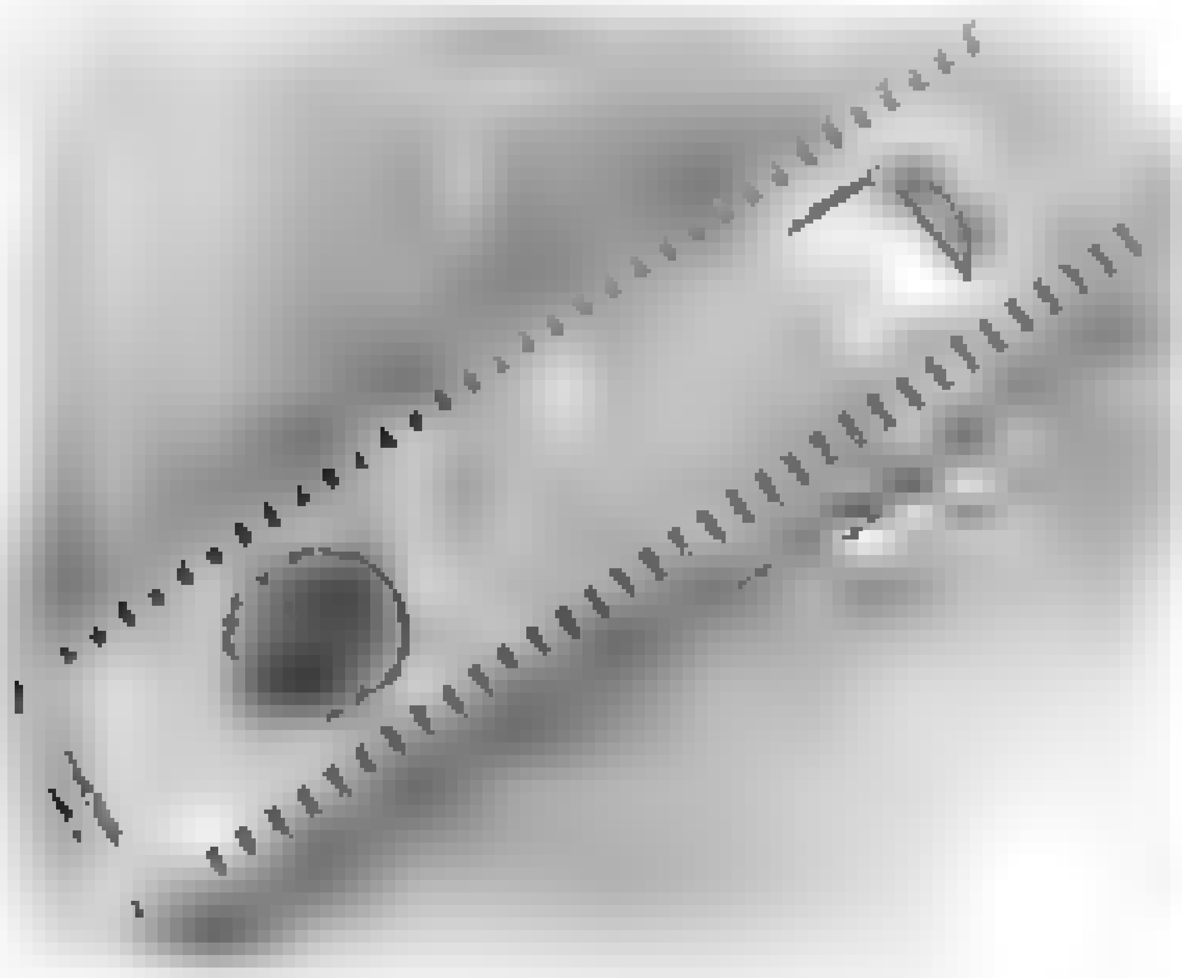
There are other aspects to a computer being user friendly. As mentioned, if you want to wire the thing up yourself, or call on the local auto electrician with no knowledge of this sort of stuff, then you will want extensive, totally accurate and easy to understand documentation supplied with the ECU. This is where a mid-range unit such as the Haltech has no peers. The Haltech as far as I know is also the only unit capable of operating the idle speed stepper motor in GM and Toyota vehicles. In addition for those modifying turbo engines it has pulse modulation boost control and turbo anti-lag control as standard features.

In spite of all the advances that have been make with aftermarket ECUs we have to realise that none will do all the things that the standard ECU fitted in today's cars will do. In general the aftermarket ECU is designed mainly with ease of programming and engine management as the priority. So in a road car how do we keep the security system functioning and the air con climate control working, while at the same time achieving the injector and spark timing that our modified engine demands? Basically the answer is you can't unless the stock ECU is retained.

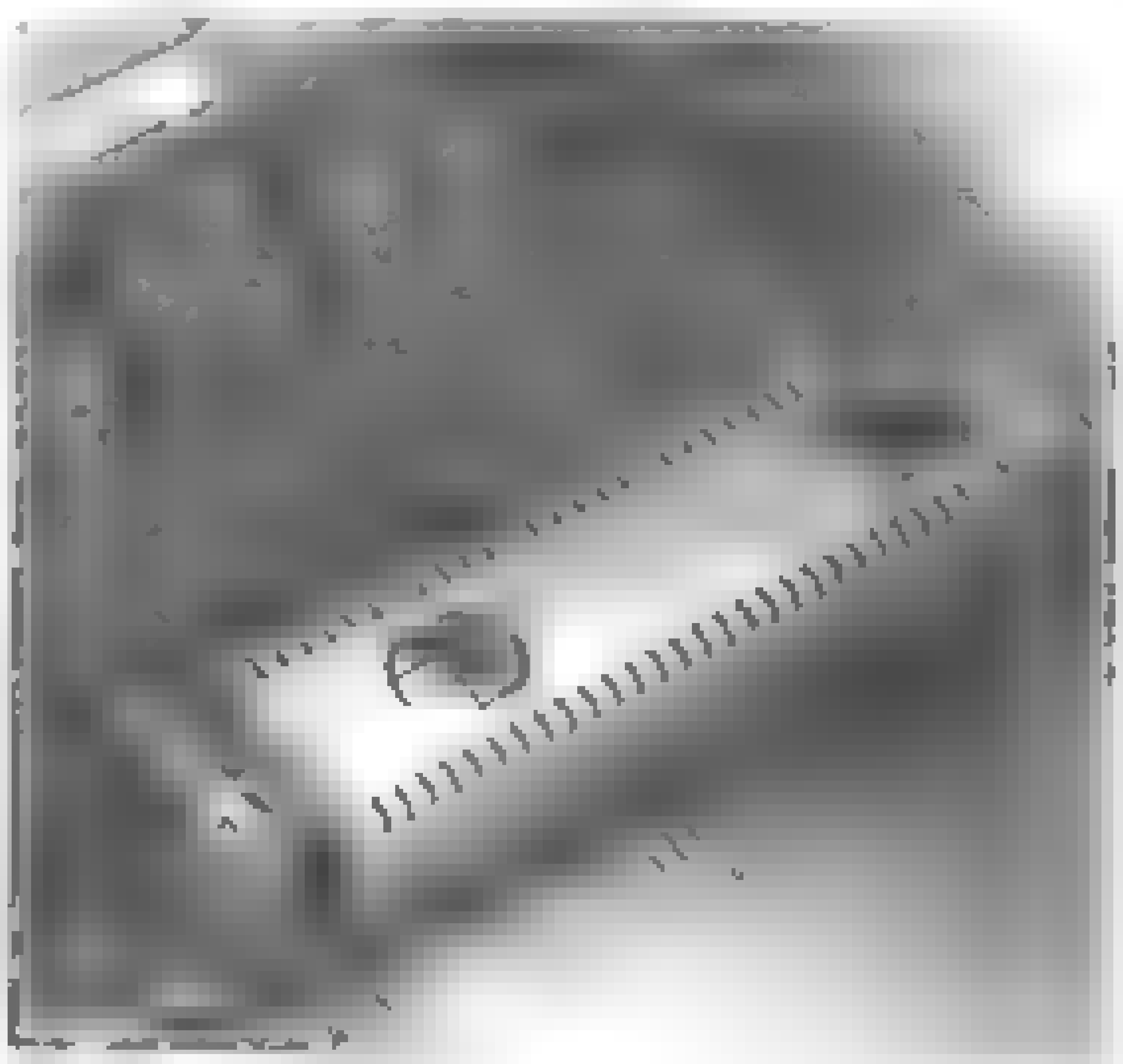
REPROGRAMMING THE STOCK ECU

The very first option we should always carefully examine is the possibility of reprogramming the stock ECU. In the past it was popular to re-chip the ECU; either swapping the EPROM chip for a chip with a different programme or else by soldering a new chip onto the stock ECU circuit board for those computers which didn't have a plug-in chip. Usually these replacement chips were supplied by mail order, and at huge expense. As such they didn't do much for performance, not to mention engine life and fuel economy, in spite of very impressive advertising claims. The truth is that tuning of this sort can only be done on the dyno. Even a dozen precision assembled race engines of the same specifications, put together by a top engine builder, will require slightly different ECU calibration to deliver optimum performance, so how can a programmer burn a perfect chip for an engine he has never seen? The truth is that he can't, he is only guessing.

The EPROM chip can be reprogrammed to adjust many parameters affecting engine performance



The chip plugs into ECU socket as shown



However, if done properly on a dyno, using good software which enables the

plus it should pass emissions tests

RUNNING TWO COMPUTERS OR A PIGGY-BACK SYSTEM

on how far we have gone with our modifications

For example we are not limited in how far we can go with our modifications, and when we bring the car back to stock, or if we resell it, it is very easy to remove the

injection, ignition timing, boost etc The stock ECU will retain control of all the other functions

There can be problems though with the two ECU system when there is a

is one problem area, as is electronic throttle control and electronic clutch arrangements

very capable unit would appear to infer that by its name - Unichip

distributors as part of an install and map package

An interceptor is cut into, or plugged into, the wiring harness to receive and then modify signals prior to the stock ECU receiving them. (Some systems modify the signals going out from the ECU, while others modify a few of the incoming signals as well as some outbound signals.) As a result of receiving input that has been [illegible] with, the stock ECU will then send out a different set of fuelling and ignition advance instructions. And if the stock installation features a pulse modulation boost control valve that setting can also be modified. Cars with speed limiters and over-boost cut out can also have those functions disabled.

However, the interceptor has its limitations which is why if it is possible to reprogramme the stock ECU or run twin ECUs that would be the way I always prefer to go with road cars. What has to be understood is that the interceptor is limited by what the manufacturer has put on the stock maps. It can only replace one set of tuning codes with another bundle that are already available on that particular map. Consequently an interceptor is only suitable for reasonably moderate engine modifications. Usually ignition timing isn't a worry as most allow ±12-15° from the stock settings, but fuelling is a problem as you will often be limited to not more than a 15–20% increase and it usually isn't possible to change fuel pressure or fit larger injectors as many stock maps don't have sufficient latitude to permit the interceptor to lean the fuelling off the required amount when the engine isn't at wide open throttle and full boost.

However, if there is no alternative but to employ an interceptor then we have to accept the limitations that go along with that course. This means choosing the best interceptor for the job at hand and then taking care to keep our tuning efforts within the bounds of what the interceptor can comfortably accommodate. Very often the reverse occurs with the enthusiast modifying the engine and then being disappointed by the final result because the engine spec is way beyond what any interceptor can handle.

PROGRAMMING THE COMPUTER

Regardless of the path we have chosen the end result will only be as good as the steps we have taken to ensure that our engine management system is correctly programmed. Thus enabling our engine to produce maximum power along with good reliability, and also good fuel economy if that is of importance. In essence we have to be prepared to put in the time running on a dyno to get the right setup. That might sound simple enough but it is anything but. The big problem is that there are lots of tuners around who don't have a clue about mapping, and equally there are many dynos about that are completely unsuited for mapping.

A new ECU and the mapping of it represent a huge investment for most people, and I'm not just talking in terms of the cash outlay involved. Really it doesn't matter how much money and effort you have put into the engine and the rest of the car, if the engine management system isn't spot on the whole exercise has been pretty much wasted effort. Therefore it is imperative that you choose carefully. Don't just think in terms of buying an ECU, or a chip, or an interceptor. Rather plan on buying a package that includes suitable hardware and software, plus a tuner who understands and can get the best out of that hardware and software, and who is working on a dyno suited to mapping.

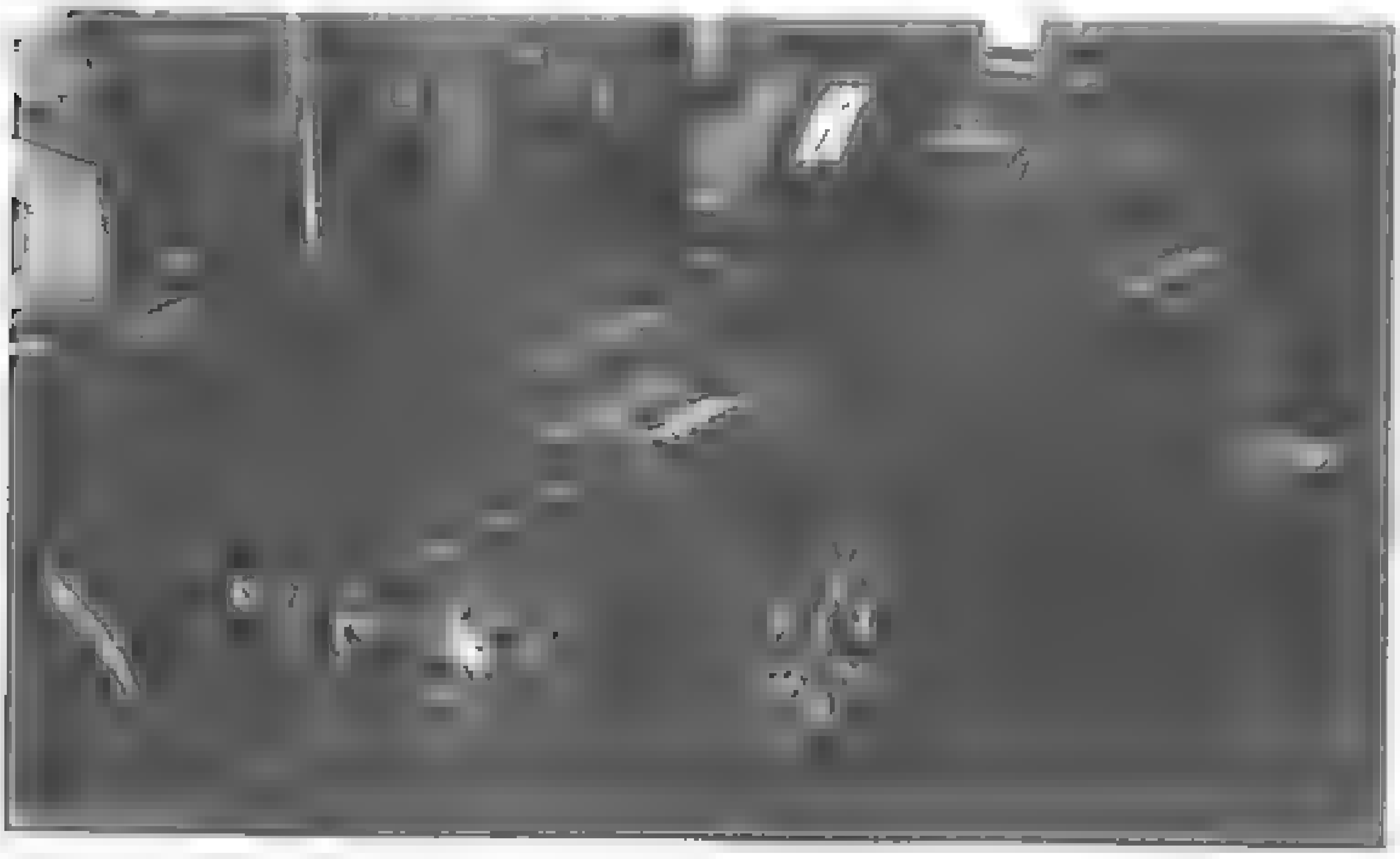
So often I see enthusiasts pay good money for rubbish results. They buy the I C think they need and then try to find someone who can map the thing. I real t is very much a catch twenty two situation, we are dealing with relatively new and constantly evolving technology here, and every tuner has to learn somewhere. Th okay if he is learning in his time and at his expense, but when you are pay s not a great idea d invariably find that th ated several times with yo one tuner to another before y have the car running satisfactorily, much less the way you know it should

CHOOSE THE RIGHT DYNO

When it comes to developing engines the engine dyno is always the way to g ver mapping is different. Ge ferable for mapping to be d chassis dyno – a rolling road. This ly so in the case of turbo e under bonnet heat has a huge influence on engine tune. Obviously you will m this statement that the bonnet should be closed for final map fine tuning ly replicate the actual environment that will exist on the road or track.

However, not just any rolling road is suitable. What we need is a modern current electronic dyno that allows the operator to hold the roller speed. Many newer dynos are the much cheaper inertia type, with the Dynojet perhaps the most w known. Instead of having an electric brake or a water brake these h roller (2½ tons in the of the Dynojet) which the tyres accelerate. Calc e at which the roller accelerates, the dyno computer can then do a print out of th vehicle's hp at the wheels. That's fine if your only concern is determining th

Any engine management system is only as good as the time and effort ping. This is best done by an operator who knows his stuff



power while accelerating at full throttle as the test is very fast and the likelihood of engine damage is minimal. However, when properly tuning an engine, we are not just interested in how it runs from low rpm to maximum revs at wide-open throttle. We want to get it accelerating well without a stumble and without detonation over a broad range of engine speeds and a wide range of throttle openings. Additionally we want the engine to be capable of holding maximum engine revs at full throttle for several seconds just as it will be called upon to do lap after lap on a race circuit. (In [illegible] easy to go overboard with excessive boost and spark advance, or a too lean fuel [illegible]

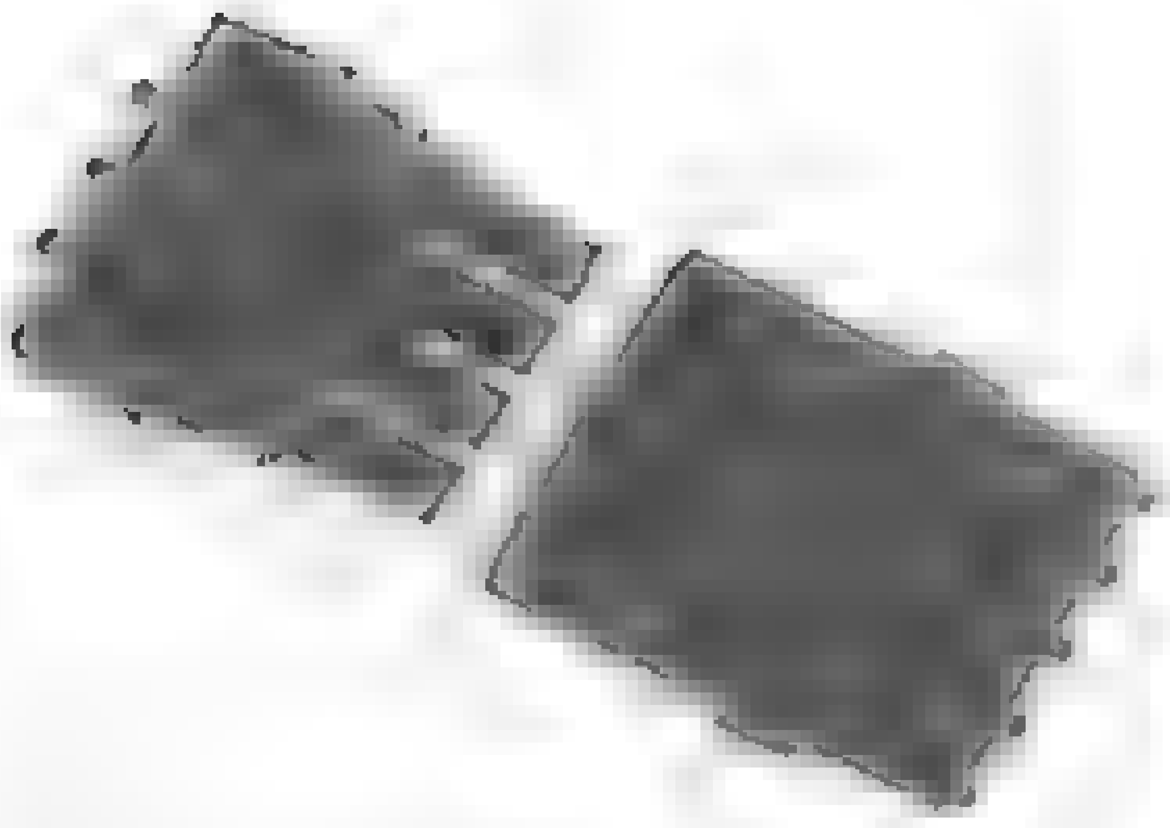
The eddy current dyno, when switched to closed loop mode, has this ability as an [illegible] calibrate the ECU with the engine held at a fixed engine speed. Having dialled in the rpm he wants the dyno to hold, he can then set about tapping in different fuel and ignition settings and the engine speed will not change – only the torque readout will change, indicating whether the new settings are to the engine's liking. When happy with the result at that rpm point and throttle angle the operator can move on to the next site until the map has been completed.

A similar thing can also be accomplished with a water brake type chassis dyno, but it is more of a fiddle and takes longer. Like the eddy current type the water brake can be used to map at any throttle opening, but the problem is that as the spark or fuel is altered and power output goes up or down engine revs will rise or fall in response to those changes, moving the engine to a different rpm and load site on the map. The operator then has to adjust the water brake to alter the load on the engine to get back to the map target site. Naturally this takes time and costs you money. Plus there is always the risk of the operator becoming frustrated at the engine revs constantly jumping about and simply giving up before the best setting has been established. This matter of the engine rpm changing is especially troublesome at the instant high output turbo engines come on boost, and of all engines they are probably the ones we need to be extra careful with to ensure that the mapping is spot on.

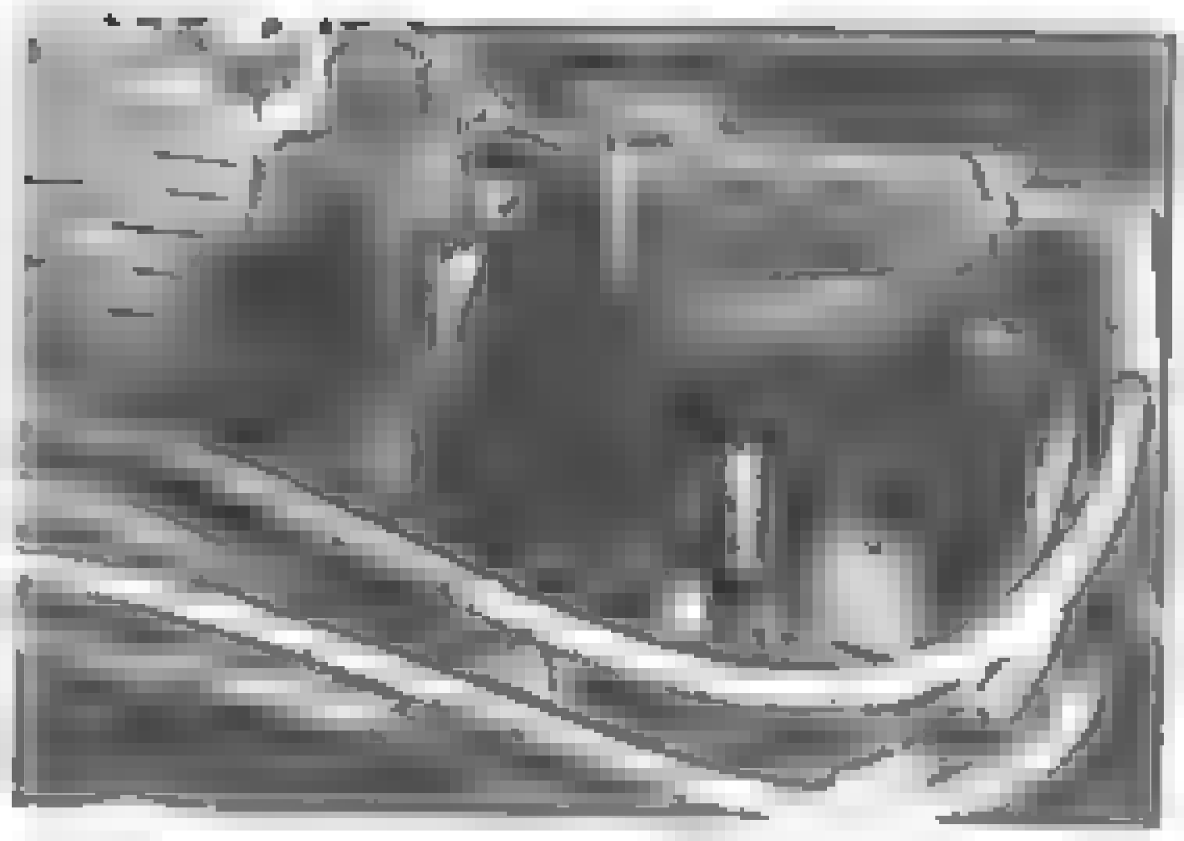
ENGINE MANAGEMENT SYSTEM INSTALLATION

Dyno operators are only human, so if they see sloppy component installation, bodged up wiring etc you are not likely to leave with a satisfactory mapping result. Either he will lose heart and give up pretty quickly, or he will reason 'this bloke doesn't care, so why should I?' Consequently do all you can to do the job correctly. The finished result should look neat and tidy.

All joints should be properly soldered and insulated. All wires should be tied to [illegible] appear okay, but inside the conductors can break, making contact sometimes and not at other times. Make sure connectors can't pull apart and are waterproof so as to avoid problems later. Remember we are dealing with tiny electrical flows in many parts of this system, so we want to prevent corrosion or poor contact at all connectors and terminals. For this very same reason we need to very carefully route all cables. If the instruction manual says to keep certain wires apart then do so even if the job doesn't look as neat. We don't want electrical current in one wire producing current, or



Use good quality waterproof connectors when wiring in the ECU and any associated sensors



This system clearly states that additional wires should not be routed close to the distributor. Doing so could alter the triggering/speed sensor signal

electrical 'noise' in a neighbouring wire that the ECU may read as an instruction from a sensor.

Take care to correctly earth shielded cable too. The shielding, a braided layer of copper or aluminium wire between the inner and outer plastic insulation, can only shield the conductor in the middle of the cable from induction or noise if it is actually earthed. That means paring back the outer insulation, twisting up the braided shielding wires, soldering on an earth connector, and then attaching that connector to a true earth. A big lump of metal, if it doesn't connect back to the battery earth, is not a true earth. Needless to say neither is plastic, fibreglass, or carbon fibre. Even major car manufacturers can get this wrong. A friend is a Philips technician. Their radios are fitted by many different car manufacturers, but in one particular brand reception and station range were terrible in certain cars. The radios worked fine on the bench, back [illegible] searching for the real rogue. It turned out to be a false earth. Up under the dash there [illegible] of being metal; the radios were being earthed there.'

The major sources of electrical noise are the spark plug wires and spark plugs. If you are from the old school it is hard to accept that tossing away perfectly good, highly conductive solid copper leads, and spark plugs without resistors or resistor gaps, can be a good thing. However, believe me they have no place in an electronically managed vehicle. Fit high quality suppressor leads and resistor plugs and don't fre Your new, powerful ignition system will give a fatter spark – even with these seemin impediments – than anything possible from old systems.

Many problems can be avoided if you first carefully read the manual supplied with the ECU. If the instructions are not clear, talk to the maker. As I have previously mentioned, if the instructions seem to make sense on a quick read-through, re-read them as no instructions are that easy.

Some things are pretty obvious but should not be taken for granted. The ECU doesn't like heat, water or excessive vibration, so mount it appropriately. Remember

ECUs are upset by electrical noise. Spark plugs and cables are a major source of th... terference, so resistor plugs, iden...ed by the 'R' in both NGK and Champio... should be used.

... be cool at cruise but turn into a stove-top at continuous full throttle. If th... ECU has to be insulated from vibration by mounting on rubber pads then don't tight... screws down so that the pads become s...

SPEED/CRANK ANGLE SENSOR

... you are sticking with the original electronic distributor you only need a few simple checks. Firstly, make sure the new ECU can read the type of signal the distributor leavers. Some ECUs can handle many different signal forms, others are quite specific. Secondly, check the mechanical integrity. If the shaft and/or bearings are worn ...owing ... shaft to wobble, this needs to be fixed. Or if the distributor has a helical drive gear ensure that thrust washers of the correct thickness are fitted to keep end float down to about 0.003in. Remember the ECU is taking its cue from here regardin... crankshaft angle and piston position in the cylinder, so if the distributor shaft is ...lowed to ride up and d... ...r wobble then that inform...n will be incorrect ...ly, if there is oil in the distributo... ...tributor oil seal ...o ...kcase blow-by pressure is forcing oil into the distributor spend your mon... ...ning th... engine first!

When ...ng from a mechanical distributor with bob weights and vacuum ...ce mechanism to an electr... ...ts type, you should first check if a l...te model of your engine utilised e... ...management and a distributor that can be easily adapted to your engine. If so, and your new ECU can decipher its s... ...distributor from a wreck and follow the three steps in the previous paragr...

If there is no easy alternative but to stick with the bob weight distributor there are a number of other things to take care of, in addition to the above three. Obviously the bob weights can be removed and the shaft welded. Likewise for the vacuum advance [illegible] it simpler to use a toothed triggering wheel and pick-up from a later distributorless engine and use the distributor only to direct the spark to the correct cylinder.

[illegible] Moving the plug wires around the distributor cap won't fix it. Under load the rotor may be close enough to the correct terminal to fire the correct plug. However at cruise [illegible] physically, to the adjoining distributor cap terminal, and so fire the wrong plug.

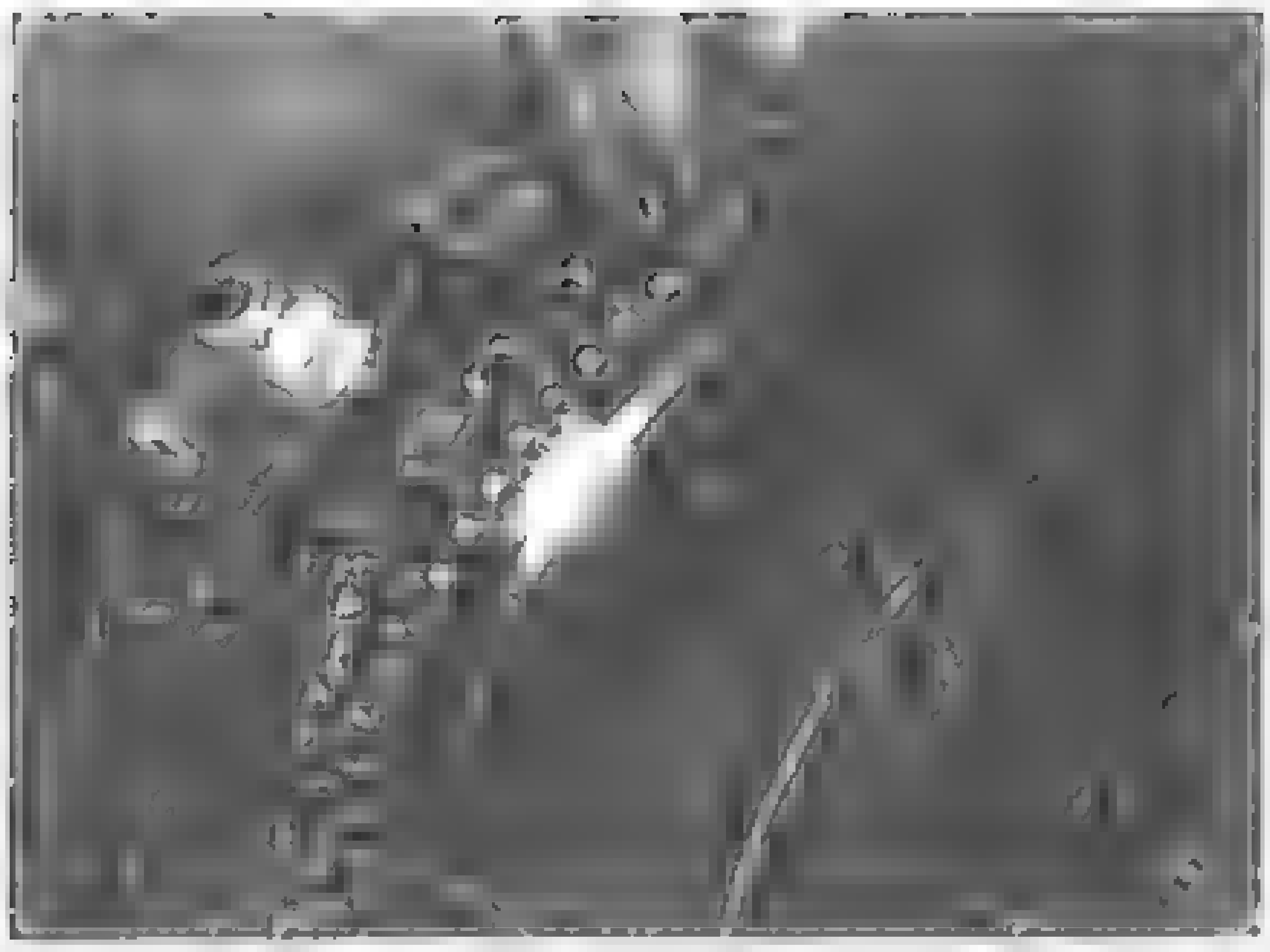
The way around the problem obviously is to reposition the rotor button on the distributor shaft. How much it needs to be moved, and in which direction is another problem. Ask around, if it's a distributor that is commonly used for engines with [illegible] may be able to find the answer on the internet.

If you can't you will have to leave sorting it out until you tackle mapping the system. Before that you can save time and ensure better accuracy if you remove the [illegible] located, paint corresponding marks on the distributor body. Later the dyno [illegible] is the same as that indicated on the ignition timing map. If they are different the [illegible] ensures that the on-screen ignition advance numbers reflect the actual amount of advance when the plugs fire.

Following that remove the distributor cap, and with No 1 cylinder on the compression stroke, slowly rotate the engine. Particularly in the range of 10–60° before TDC [illegible] scrape the distributor cap around one side. If it is located by a slot in the shaft lift the [illegible] If you use a hacksaw you shouldn't have any trouble identifying the correct slot to use. If you are more fussy and mill the slot be sure to file a 'V' or put a hacksaw cut in the [illegible] cap and put a paint mark at No 1.

TRIGGER WHEEL SYSTEM

The alternative, as explained in the previous chapter, is to run a triggering wheel on the crankshaft along with a fixed pick-up. Here we can choose to use an expensive



Locating the trigger wheel on the drive end of the crankshaft improves timing even more than that possible with it on the pulley end. Plus, being located in the ... using, the pick-up is not exposed to damage from flying track debris or drive ... However, in the event of sensor failure this can make replacement more difficu...

triggering arrangement such as that made by MSD for all-out race engines. Th... features an aluminium wheel with a number of rare-earth magnets that rota... past a non-magnetic fixed pick-up to send a triggering signal. Another arrangemen... suitable for high rpm running is to use magnetic studs in the crank pulley or flywheel. As these pass a magnetic pick-up a triggering signal is transmitted.

The relatively inexpensive way is to utilise something from one of th... makers. When scrounging for a trigger wheel ... a pick-up, but prior to doin... be sure you know what wheel and pick... you need to communicate with your ECU. Most of these systems are reliable, and obviously will ... the job done at normal engine rpm. However the 35 tooth (36–1) pressed trigger wheel Ford use on some ... can be prone to s...ding or bending teeth at continuous high rpm. Shorten... ...ut cut off no... take care when strai...ening t... ...r you don't ... off the fracturing ... ...urself. The Ford wheel should now live at up to a regular 7,500rpm. The 58 tooth (60–2) Bosch t...ger wheel is mo... ...to find in wrec...

...ng on a location for the pick up get a good idea of how you ar... fix the ...ooth wheel to the crankshaft. Don't mount it at this time, just have a... where and how you will fit it up. With that figured out find the best location for the pick-up sensor. You want it in a p... ...n't be vulnerable to dam... ...re it is not going to make normal service ... a lot more difficult. Mount... ...and allow for adjustability. If the ECU has no provision for a few degree... misalignment of sensor and trigger wheel you will have to use slotted mounting ... to get the TDC reference correct. Also it is a good idea to provide for a... ...nt between the teeth and pick up sensor. If the signal is a bit weak or suffer...

...at noise, moving the sensor closer can sometimes help. Conversely ... nning the Ford wheel at very high rpm you may want to make the gap wid... e risk of centrifugal force spreading the teeth and taking out the ser...

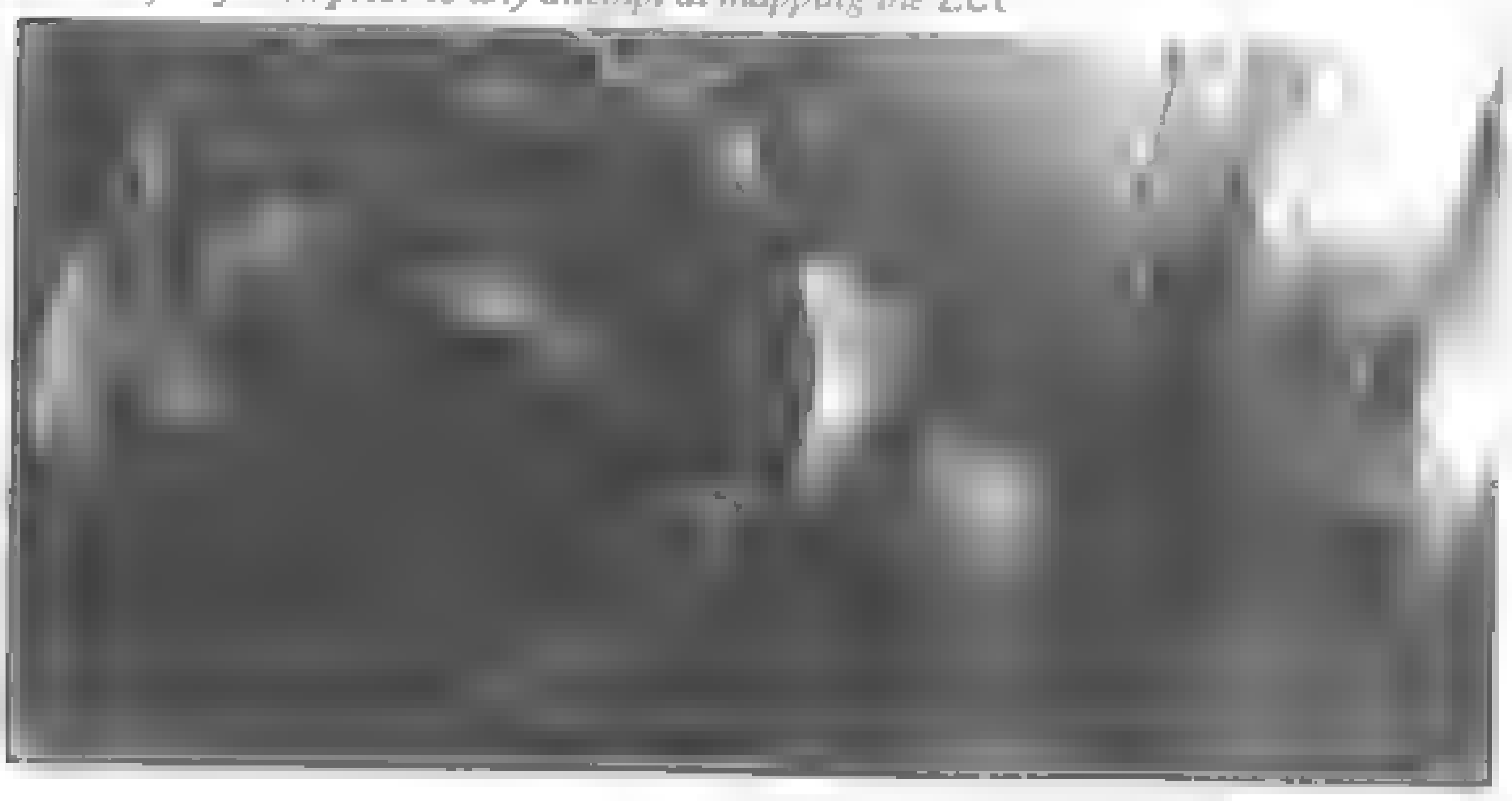
After this the tooth wheel can be properl... d in position. Here you will requi... e information from your ECU manual as to the triggering angle before TDC. It states 120° before TDC then you will have to very carefully move the crank to t... angle on cylinder No 1, and lock it there. Then fix the trigger wheel so that the first tooth immediately after the gap is lined up with the pick-up. How this works is that t... ECU receives regular pulses as each tooth ... s the sensor. Then when it misses pulse because of a missing tooth it kn... s the crank angle before TDC. Th... ...ediately after the gap is its reference point. If that reference angle isn't s... then all of the ECU's calculations will be erroneous by that angle of error to...

Some people are confused about how many teeth work best when ... gnetic stud per two cylinders seems to be fine on many competition ... Basically more teeth keep the ECU more regularly updated as to cr... shaft speed nothing more. Niss... n chose to inform their ECU of crank rpm every 1°; Bosch with their 60–2 wheel decided on letting the ECU know engine rpm every 6°; most ... have gone with a 36–1 tooth wheel, meaning the ECU gets a new crank rpm ... every 10° of crank rotation. Many competition systems get the latest c... k speed inf... the ECU as infrequently as every 180° (4-cylinder engines), 120° (even fire ... cylinder), 90° (V8 engines). The c... by some is that more frequent updates make for a sharper, more responsive engine. I must be missing something, but then I chose to b... a tu... er rather than a racer, so what w... ld I know.

THROTTLE POT SENSOR

Along with some fuel injection systems, carburetted engines with electronic en... ...ment also require a throttle pot to inform the computer of how wide th... throttle

The throttle position sensor informs the ECU how far the throttle is open so must be ...fully adjusted prior to any attempt at mapping the ECU



is open. For the pot to properly inform the ECU of the throttle position you have to set it up properly. First, ensure that the throttle is opening fully. Get someone in the car holding the pedal to the floor and have a look. This is important, because if you get half-way through mapping and find you can't fully open the throttle and you then set about rectifying the problem all the mapping up to that point is lost!

With full throttle taken care of, back off the throttle screws to allow the throttle to fully close. If there is any oil, carbon, dust, or grit in the throttle body or on the butterfly/s, get it cleaned out. The throttle must be allowed to close fully before you set the throttle pot closed position. Now fit the pot over the 'D' on the end of the throttle shaft. Don't force it into the fully closed position; you want the pot just fractionally open. When all is as it should be tighten the mounting screws.

PRELIMINARIES TO MAPPING THE ECU

This section is included to assist those tuners who have access to a dyno and would like to learn the noble art of ECU mapping. As previously stated you need an electronic eddy current dyno – either engine dyno or rolling road. Don't even momentarily entertain the idea of mapping on anything else. Start our learning with an engine that's already run-in and that isn't turbocharged, supercharged or has wild valve timing. An engine in good mechanical order, running nicely on Weber carbs and some type of mechanical ignition, either points or electronic, and is now being converted to fuel injection and electronic ignition, would be a good practice tool.

Before you even attempt to start the engine there are a number of basic checks that are necessary. Don't fool yourself, mapping isn't easy, so anything you can do to minimise frustration along the way will help you to achieve a good outcome. The battery must have plenty of cranking capacity to spin the engine fast, and have sufficient voltage to give a good fat spark so that you have some sort of sporting chance of firing a fuel mixture that is likely to be excessively rich or massively lean.

If the car didn't come from the factory with fuel injection, your first check should be to ensure the fuel supply system is working. Connect a fuel pressure gauge into the fuel rail and turn the ignition on. You should be able to hear the pump whirring for a few seconds getting the system up to pressure, and then stop. In-tank pumps have their noise muffled pretty well so this may not be so obvious. Take a look at the pressure reading. It should be holding steady, and in the case of a fixed pressure regulator, be fairly close to its preset pressure. If not and the pressure is very high have a look to ensure that fuel hoses into and out of the pressure regulator are connected to the correct ports and have not been reversed. A low reading could indicate that the regulator has dirt or grit in it, so blow it out with high pressure air in the direction of flow, and try again.

With that done it's time to link your laptop to the ECU. Select the live mapping tab (it may also be called live adjustment mode, live communication mode etc), and take a look at the screen. Here common sense comes into play. If some readings don't look right then it's probable that there is a problem somewhere, so don't ignore it and assume that it will go away. For example the engine isn't running as yet so the rpm should be 0, the engine water temp and air temp readings should be pretty much the

same etc. Any readings outside of what you consider normal must be followed through by checking that your connections and wiring are correct, or that sensors are working properly; yes even new ones can be faulty.

If the engine has a throttle pot you most likely will have to adjust it in [illegible] the maximum stated in the manual when fully open.

Now it's time to begin cranking the engine over. Murphy's Law dictates that it [illegible] feeble, the injectors may not open, and the ECU may not compute.

Next determine if the lack of fire in the cylinders is ignition – either advance [illegible] or is it fuel – too lean or too rich. There are two easy methods to check this. One is to spray a starting aerosol into the inlet tract – not the air filter – as the engine is cranked [illegible] inlet manifold squirt the fuel down each inlet tract.

If the engine now runs, even if only for a few seconds, you know the ignition is [illegible] smooth idle. When it is reasonably smooth set it at 1,150–1,300rpm.

If the engine doesn't run when the aerosol or fuel is sprayed in, instead just [illegible] backing off the idle screw to bring it down around 1,150–1,300rpm.

Before you go any further you now have to carry out an ignition advance [illegible] reference angle to be corrected, or it may be necessary to physically move the distributor, or pick-up in the case of tooth, or magnet, wheel arrangements.

SORTING IDLE MAPPING

With the preliminaries taken care of we can get into the serious business of getting the [illegible] don't want more than about 10° at idle. Sure the engine will rev a lot faster with [illegible] not impossible, to get the fuel map right. With stacks of idle advance the throttle

plates need to be almost totally closed to keep idle speed down to 1,000rpm. Consequently with very little air going into the engine it only requires a correspondingly small amount of fuel. Injector flow is pretty erratic at low flow so mapping gets quite tedious and idle quality will suffer.

If you are getting spit back in the inlet system it probably means the mixture is [illegible]

slight decrease in injector flow causes a big, sudden drop in idle speed as well. Now with low air speed in the inlet tract a lot of that fuel will drop out of suspension, the engine will go lean and may even stall. Another scenario is that even if you manage to get idle fuelling half decent the engine will experience a massive off-idle flat spot that will make the car a big problem to get off the line unless lots of throttle and clutch slipping are brought into play. These hassles are avoidable: just use 10° idle advance for a street engine.

Below idle though the story changes. At the 500rpm speed site you can give the engine a heap of advance, often up around 30–40°. With all this advance below the 1,000rpm idle speed site, any time the engine speed drops below 1,000rpm more advance is added, nudging the idle speed back up. This provides for a smoother idle and virtually eliminates any chance of the engine stalling when the air conditioner or radiator fan cuts in while you are stopped at traffic lights.

The situation is similar with regard to fuelling. Below idle the fuel number has to get bigger simply because if the engine begins to lose revs air speed falls and a lot of fuel will fall out of suspension, causing mixture leanness. Therefore if the fuel number at 1,000rpm is 30, then make it around 40–45 at 500rpm and 55–60 at 0rpm to ensure the mixture is actually combustible.

IGNITION MAPPING

With the idle sorted we can move on to roughing out a base map for both the spark and injectors. If the engine was previously running well on a bob weight distributor then those ignition advance numbers will be a good starting point. If you don't have any idea where to start then it pays to be conservative to begin with; you can always give a working engine more advance later.

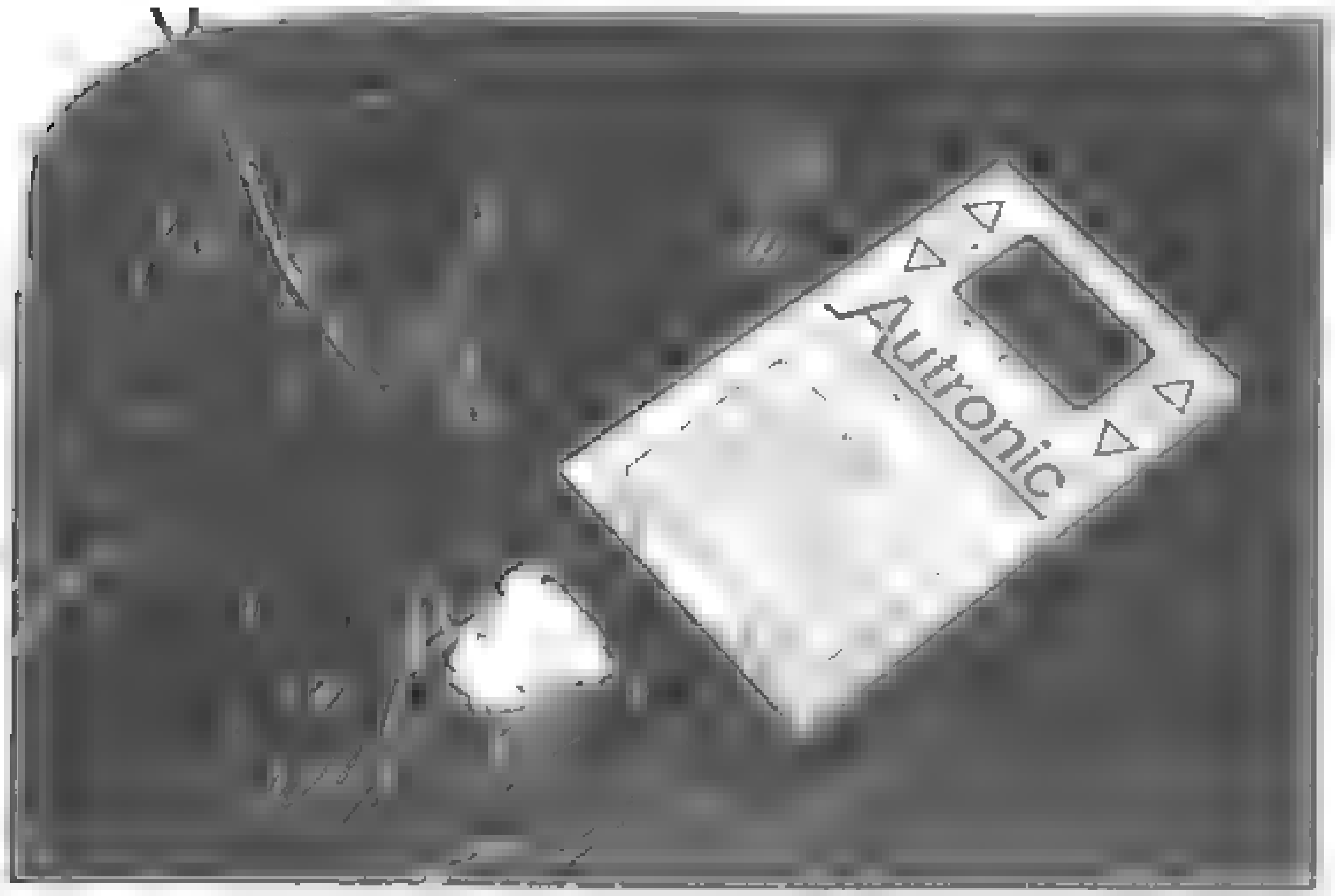
Table 8.1 will provide a starting point, but remember that I don't know what engine you are running, nor its state of tune. Nor do I have any control of how lean you have gone with the fuel mixture. Consequently if you don't have a warning light or alarm connected to a knock sensor to alert you to detonation take care to listen out for it and don't focus completely on how the engine is performing! Conversely don't fool yourself into thinking that grossly retarded ignition advance won't cause any harm. Late ignition means a lot more heat in the exhaust passage. That means hotter exhaust valves and seats, and exhaust headers. Excessive heat here could see the valve seats cracking, distorting, or dropping out of the head. The exhaust valve head could begin to melt, distort, or even fracture and fall into the cylinder. An exhaust glowing red is a warning; it may look spectacular, but don't ignore this sign. Additionally, mapping while retarded, it will be a lot more trouble to get the fuelling correct. With retarded ignition the engine may respond favourably when an excess of fuel is available. Later when you realise the advance is way too retarded, you will have to waste a stack of time rewriting the heavy load fuelling a good deal leaner.

Table 8.1 Generic ignition advance map

		Ignition Advance	
	Bore dia (mm)	2,500rpm	4,000rpm
[illegible] full throttle	100	25	32
	[illegible]	[illegible]	[illegible]
	[illegible]	[illegible]	[illegible]
[illegible] quarter throttle	100	3[illegible]	[illegible]
	[illegible]	[illegible]	[illegible]
	[illegible]	[illegible]	[illegible]
[illegible]ive full throttle	100	2[illegible]	[illegible]
		18	[illegible]
	[illegible]	[illegible]	[illegible]
[illegible]e quarter thrott[illegible]	[illegible]	[illegible]	[illegible]
	[illegible]	[illegible]	[illegible]
	[illegible]	[illegible]	[illegible]

Note: use the suggestions in this table only if you do not h[illegible] [illegible]ance information for your particular engine. These figures are a good st[illegible] [illegible] for engines with alloy cylinder h[illegible]ss[illegible]on ratio of 10.1 [illegible] temperature 80°C or less, burning 98 RON summer blend unleaded fue[illegible]

[illegible] *Lambda probe w*[illegible] [illegible]*6.0 1. Map*[illegible]



FUEL MAPPING

With a base ignition map written, have a 'feel' for how the engine is running. The idle should be stable and the engine shouldn't be thrashing about. Now blip the throttle and at the same time watch the lambda number. Probably the engine will splutter and die. If the lambda shows you need more fuel, then add this to the load sites just above and below idle. Keep this routine going until you are able to get the engine to cleanly rev up when the throttle is blipped. It should now be possible to drive off from idle and run up over 2,000rpm.

Now we can move to the more serious fuel mapping. On light throttle, at around 2,500–3,000rpm, adjust the fuelling until the engine runs crisply. Then move through the load sites until you are about midway between quarter and half throttle, aiming to get the engine feeling smooth and crisp.

With that done we need to take a deep breath and have a careful look at our map. The fuel figures shouldn't be jumping about all over the shop. When we look at the numbers they should be all heading in the same direction, either progressively increasing or progressively decreasing. Any number out of line with this trend should be tweaked a bit to 'smooth' the map. For example if a number at one site is bigger than its neighbours, lower it and perhaps raise the neighbouring numbers a little.

When the engine is for road use then you have to be concerned with fuel consumption at legal speed limits. You don't want the mixture so lean that the engine is surging or popping through the exhaust. Operating like that, like running rich, also wastes fuel. On the other hand don't get too concerned that going lean you will cook the engine, melt the pistons etc. At small throttle openings you won't get into this sort of trouble. Just get sufficient fuel in to avoid surging and exhaust popping, and enjoy the smaller fuel bill!

Competition engines are different in this regard. Usually we simply want them to run well at lower rpm so as to get on the cam as quick as possible. However, if fuel consumption when behind the safety car is an issue then we need to factor that in.

At these lower engine speeds and smaller throttle open angles, don't be too concerned with lambda numbers; simply aim to get the engine running nicely. As we move past half throttle though the situation changes. Now we have to pay a lot of attention to the lambda. Quickly blip the throttle and note how the engine responds. If it stumbles and the lambda went lean, add fuel to those parts of the map. Conversely if it reads rich take fuel out. There is no point in taking the engine to max rpm at this stage. Most engines need the biggest fuel number at peak torque revs. Therefore if you know the engine makes best torque at 6,000rpm don't waste time now mapping at higher rpm.

Keep at this until you are happy with how the engine feels. When you reckon the fuelling is close and that you won't damage the engine when doing more severe testing, it's time for you to take another look at your fuel map. Again search for numbers that are out of line with their neighbours. When such are found then smooth the fuelling by changing the target number, and follow up with a mild tweak of the

peak torque speed. From peak torque right up to the rev limit simply put the same fuel numbers in and leave tweaking top-end fuelling until we do a full power curve.

Run the engine again and double-check that at each site above about half throttle the fuel number is safe with an air fuel ratio around 12.5:1–12.7:1. Don't get too

carried away striving for a particular number, anything in that range will be safe. Just be sure to keep a close eye on the lambda and don't hold the power on and make fuelling changes. This is hard on the engine, particularly at anything past 90% throttle and at rpm past peak torque. Always get off the throttle and only then adjust the fuel number. When you are absolutely sure that all is in order we can proceed to the exciting stuff – producing a power curve!

The idea of course is to produce the highest power at each rpm/load site. What further complicates things is that several fuel numbers will give the same hp. Remember the dyno is an artificial environment. In a real environment an engine accelerates through a number of speeds; engine loads change as tyres grip up and lose traction; the vehicle is moving, so air flows around the engine are in a state of constant flux etc. These factors, and many others, affect the temperature of the combustion chamber and piston crown. So in the real world what fuel number should we choose? There is no simple answer to this, but as a general rule I would go for the low number at 50% up to about 80% throttle. Above that I incrementally increase fuelling so that I'm about midway between the high number and the low number by the time I get to 100% throttle. If I'm tuning for endurance racing and the track has only a relatively short straight and no steep climbs, the fuel number may possibly be safely trimmed without risk of engine damage, and incur only marginal power losses. However, if there are big climbs and/or long straights I am more inclined to go closer to the top fuel number above 95% throttle. This will help the pistons live and will keep the hp up for the full length of the climb or the straight, hopefully compensating for less track mileage between fuel stops.

With the oil and water at operating temperature, do your first acceleration power run. Don't take your eye off the lambda meter for an instant. If at any point the lambda goes lean abort the run, add more fuel, and try again. When you manage to do a full acceleration run with good lambda numbers stop and take time to examine the power curve. Find at what rpm peak torque occurs. Remember previously we only guessed this. Now add a little fuel at rpm past true peak torque up to the rev limit and do another full acceleration run. Did top-end power rise? If so, it tells you that you may be losing fuel out the exhaust during the cam overlap period and/or the exhaust tuned length could be a problem. Then again there may be no clear answer; some engines just demand more fuel. If adding fuel wasn't to the engine's liking try pulling a little out and do another power run.

Now overlay the three power curves. You may find that a dip in one curve didn't exist, or wasn't as severe, in another curve. If that's the case then change the fuelling at that site/s and do another acceleration run and again overlay the curves to confirm that you are heading the right way.

IGNITION MAP

With the base fuel map in place we can set about constructing a proper ignition map. This is fairly straightforward. Just remember that as for fuelling a range of spark advance numbers will give the same hp. However, unlike when building the fuel map this time we always use the lowest advance number. Additional advance over what gives max hp never helps. In fact, on-track testing may reveal that you need to knock off a couple of degrees at a few sites when the piston crown and combustion chamber

are fully heated up. This, along with a touch more fuel, could be necessary on circuits where you are at 100% throttle for a lot of the lap.

Start out with mapping at light loads, and then progress through 50%, 75% and [illegible] with their neighbours.

When you are happy with the result proceed to a full throttle acceleration run. If [illegible] curve that very gradually falls past max torque rpm – and is artificially rev limited to [illegible] etc, then you may only need 3–4° more spark lead at the rev limit and slightly less, perhaps 1–3° at max power rpm. On the other hand, an engine that runs out of breath pretty rapidly at the top of the power range – indicated by a torque curve that drops away steeply – may want 5–8° more advance at the rev limit and a fairly similar amount, or maybe just 1–3° less at maximum hp revs. Don't be greedy and think adding more and more advance will help. It won't. Much more likely the outcome will be reduced performance and engine damage, plus the engine may not want to over-rev very much past max hp revs. Always strive for the highest hp with the least ignition lead.

SIMULTANEOUS RUNNING-IN AND MAPPING

It isn't a great idea to map a newly built engine until you have gained some [illegible] the engine management sufficiently sorted for an engine to run reasonably well in a short time, and then get down to the serious business of bedding the rings etc.

The basic procedure to begin with is the same as for a used engine. Follow what [illegible] straight to 2,500–3,000rpm, or higher if the cams are pretty wild. You don't want the engine running under 2,000rpm. Low rpm lubrication is poor, particularly to the valve train, so keep engine speed up.

The idea is to set the dyno load for the engine to run at 2,500–3,000rpm with the throttle on the first load site and adjust the fuelling so that the engine smooths out [illegible] revs to increase up to the next rpm site. Adjust the fuel number to get the engine running sweetly. Keep repeating this routine of reducing dyno load, moving to the next speed site and adjusting the fuel. Eventually you will reach a point where the engine won't rev any faster. From then on right up to the rev limit simply fill in the last fuel number at all of the remaining speed sites.

Now repeat the procedure with the throttle moved a bit wider open and with enough dyno load to hold 2,500–3,000rpm. Adjust the fuel, then reduce dyno load to move to the next rpm site, adjust the fuel and so on. Repeat this until you reach 50% throttle, by which stage the engine should be nicely run-in to allow you to map the high load sites and perform full acceleration runs as previously outlined under the FUEL MAPPING [illegible] next, and then map the fuelling for high loads above 50% throttle.

When the fuelling has been sorted you can then move on to finalising the base ignition map. Previously the map was pretty much an educated guess based on the figures in Table 8.1 and the knowledge you have of your particular engine; purely aimed at getting the engine running well enough, without damaging it, to enable the fuel map to be written. Now with correct fuelling we have to get the spark timing spot-on as discussed under the subheading IGNITION MAP, page 268.

ACCELERATION ENRICHMENT

With a carburetted engine an accelerator pump squirts extra fuel into the airstream when the throttle is rapidly opened. This arrangement is in place to make up for the time lag between when the throttle is floored, and all the additional air rushes into the cylinders, and when the carb is able to provide more fuel through the normal metering channels and jets to compensate for that increased air. Without the extra fuel the mixture would go lean; the engine may die momentarily and stumble with a big flat-spot.

Fuel injection basically doesn't change this scenario. The main difference is we don't have to introduce as much additional fuel into the airstream because fuel atomisation is superior with fuel injection. Plus with the injector shooting at the back of the inlet valve in the majority of engines, there is less likelihood of the fuel dropping out of suspension and remaining behind clinging to the port walls.

The main thing to watch with acceleration enrichment is that we don't fiddle about adding enrichment where it isn't needed, or doesn't work very well. I think even the major motorcycle manufacturers have been caught out here. A number of bikes were plain horrible to ride around the street at small throttle openings, constantly snapping and surging. Some of this was due to throttle bodies sized more for racing than for the street, but another culprit was improper light throttle mapping and a desire to add acceleration enrichment right from idle. In normal use you don't cruise with the engine running at idle speed and then snap the throttle open, but it seemed to me that bike makers were mapping like this. There is no point in adding acceleration enrichment lower in the rev range than the engine is ever going to be accelerated from. At higher revs the increase in air flow when the throttle is quickly opened is nowhere near as dramatic as at low rpm. Consequently the engine management and injection systems have adequate time, without adding acceleration enrichment, to get the extra fuel into the air stream before the mixture leans out.

Before you begin working on acceleration enrichment be absolutely sure that your mapping is spot-on in the light throttle load sites at engine speeds from idle up to about 2,000rpm. If the engine feels dead and unresponsive in this area the mixture is probably lean, and/or the spark may be retarded. If it's hunting and shaking it's very

rich. Don't blame the cam in an attempt to excuse poor mapping. A fuel injected engine can run quite nicely with a pretty mean cam if mapped properly.

The management system manual will inform you of how the ECU handles acceleration enrichment. Some provide for you to select a percentage increase of fuel for a time period, or for a number of injector pulses. Also there may be provision for 'decay' – how long the extra injection tapers off for, either in time or number of pulses, rather than simply switching off and returning to the base number.

Don't try to do this sort of tuning on the dyno; the road or track is where you need to be. Drive as you normally would and add acceleration fuel until there is no hesitation when you hit the throttle. Please don't expect a race engine to accelerate cleanly from ridiculously low revs when it's off the cam. And don't expect a road engine to get up and go from low rpm when the throttle bodies are way too big and you are in a high gear. The idea is to be realistic and attempt to anticipate at what rpm, in what gear, and how rapidly the throttle will be opened when accelerating in the engine's regular environment. You don't want to waste fuel adding more enrichment than the engine and your driving style demands. Also if the engine is very powerful, and traction is limited, why exacerbate the issue with perfect acceleration enrichment that leads to perfect plumes of tyre smoke and slower lap times? In many motorsport situations a big number of competitors have major problems with delicate throttle control, particularly on wet surfaces. Thoughtful limiting of acceleration enrichment can assist.

AIR TEMPERATURE COMPENSATION

There are two angles to consider here – as air temp increases the amount of air going into the cylinders decreases, so less fuel is required. You can generally figure that for every 3–4° temperature increase air density goes down 1%. Therefore the amount of fuel the engine requires will decrease by a similar amount, about 1%. However as the temperature continues to rise the engine may become more sensitive to abnormal, or even out-of-control combustion. This situation calls for less spark advance and perhaps more, rather than less fuel.

Basically the reverse occurs at low air temp. More air density calls for more fuel, but what happens with ignition advance can be a bit of a lucky dip. Some engines will happily give more power with extra advance and display incredible resistance to detonation. Others though demand less advance because the more tightly packed air and fuel molecules transfer the combustion flame faster and faster through the combustion space, until the rate is too high and the mixture explodes.

Engine management software is usually written around a standard temp of 20°C. So to protect the engine and ensure best hp, on the air temperature correction screen, there will be provision to adjust both ignition advance and fuelling to compensate for temperature changes. Unlike a few sophisticated ECUs that give full three dimensional control, most management systems offer more basic correction. Thus there will be a coarse compensation for fuelling and spark based purely on air temperature without consideration of engine rpm or throttle position. Hence if you enter 5% enrichment when the air temp falls to 5°C, then that amount of additional fuel will be added to all of the base fuel figures even though the engine may only really benefit from that extra fuel at wider throttle openings.

COOLANT TEMPERATURE COMPENSATION & CRANKING ENRICHMENT

At first glance these topics appear to be pretty similar. Provide lots of fuel when the cold engine is cranking over, and then progressively reduce fuel flow back to the base number when the engine reaches normal water temp. However, cranking enrichment is also a significant issue with hot engines, even very hot engines. Many of the problems you see, and perhaps will later experience, of hot engines failing to fire up can be traced back to cranking enrichment. Sluggish cranking rpm with race engines due to weak battery power, high compression etc can be part of the problem too. Frequently though it will be found that with wild camming, monster throttle bodies and so forth, a competition engine, even a hot one, will only fire with 20–30% cranking enrichment. The idea here is to keep adding fuel until it comes to life easily. An excess of fuel generally won't hurt provided you don't wet the plugs. Also be aware that because of the decay feature cranking enrichment is only available for a specific number of seconds. If the fire hasn't started in that time frame, don't just keep cranking, the enrichment will by now have ceased. Rather get off the starter, wait a moment, then spin the engine over again. If the plugs are getting wet you need to take fuel away, otherwise keep adding fuel until the engine consistently comes to life easily. We are assuming here that the starter motor, battery and ignition system are up to the task. Be aware though that not all hot engines want a lot of enrichment. In fact some will refuse to fire with more than 4–5% added. The idea is to commence at 10%, then go to 15%, then 20% etc. Don't waste time, 5% steps will get you close to feeling what the engine wants. With a cold engine at 15°C start out adding 60% enrichment, and don't be frightened to go past 100% if the temp is close to freezing. Depending on the software, cranking enrichment is usually halved the moment the engine starts.

After the engine begins running, injection duration and spark timing will be compensated for via the coolant temp adjustments. This can consume a lot of time as the engine quickly comes up to temperature before you have time to actually test how nicely the engine operates while cold. Mind you, this isn't such a bad thing for competition engines, purposely keeping cold running horrible so that impatient drivers don't work the engine hard until it is fairly well up to operating temperature can cut maintenance bills! For road engines you don't want the engine spluttering and dying as you enter a busy motorway. So even though running hard when cold will kill the rings and upper cylinder region the engine has to be able to take a boot-full of throttle from almost the moment it fires. To achieve this you will need long cool-down periods, even overnight, to retest your latest cold enrichment. As a guide, if you found the engine wanted 60% cranking enrichment at 10–15°C, start out with 30% at 20°C, 20% at 40°C and 8% at 60°C. If the engine feels sluggish and there is no sign of black smoke the engine is probably lean, so add fuel and try again. If it's still sluggish, pull the spark plugs; black sooty tips indicate too much fuel, nice clean plugs likely mean the engine wants more enrichment.

Many cold engines respond nicely to a few degrees more spark lead. At times though, due to emissions laws, road engines may be purposely retarded to get an unheated CAT up to temperature more rapidly. When this isn't a consideration you may like to start out testing with an additional 5° advance with the coolant at 20°C tapering down to 1–2° extra at 50°C.

MAPPING ROAD VS RACE ENGINES

A non-turbo track engine isn't d[illegible] You basically want it to start [illegible] hot, idle without splattering and dying, and then pull cleanly until it's on cam. After that it has to make maximum power over as wi[illegible] band as possible [illegible]ngles of 60–100% ope[illegible]

With road engines we [illegible] e, plus a whole lot more. The[illegible] has to be minimal fuel bur[illegible] or popping thro[illegible]h the exh[illegible] The engine has to work just as well [illegible]res between freezing and 100°C. Then to really make a tuner's life d[illegible]st be crisp and rear[illegible] to race [illegible]very conceivable throttle position[illegible]ed, even when in a gear that's [illegible]gs higher than what the en[illegible] To achieve this state [illegible]f tune takes the big manufacturers [illegible] and [illegible]lions of dollars. All we can do is an honest job and help the enthusiast understa[illegible] we need accurate input from h[illegible] [illegible]rhaps several weeks to massage away every tiny h[illegible]tation. Also if the cold engine fuelling was tackled in summer you will probably need to tweak [illegible] [illegible]richment as the mercury heads downwards in winter. Realistically if it takes hal[illegible] day to map a race engine don't be disappointed if a road engine takes anoth[illegible] [illegible]r. And the sad thing is after that extra day the engine probably won't b[illegible] [illegible]er on the dyno; it will just be a [illegible]uch nicer dr[illegible]

[illegible]out a lambda meter, cust[illegible] g feedback from o[illegible] A digital voltmeter conn[illegible]da sensor will give a rough i[illegible] [illegible]re is head[illegible]ng. If the [illegible]ports surging or hesitation at cru[illegible] [illegible]r gentle acceleration and the voltag[illegible] t 200–300 milli[illegible] [illegible]l indicate excessive le[illegible]ness. At full throttle any reading [illegible]er about 930–950 [illegible]olts should be considered lean[illegible]

TEST DRIVE MAPPING REFINEMENTS

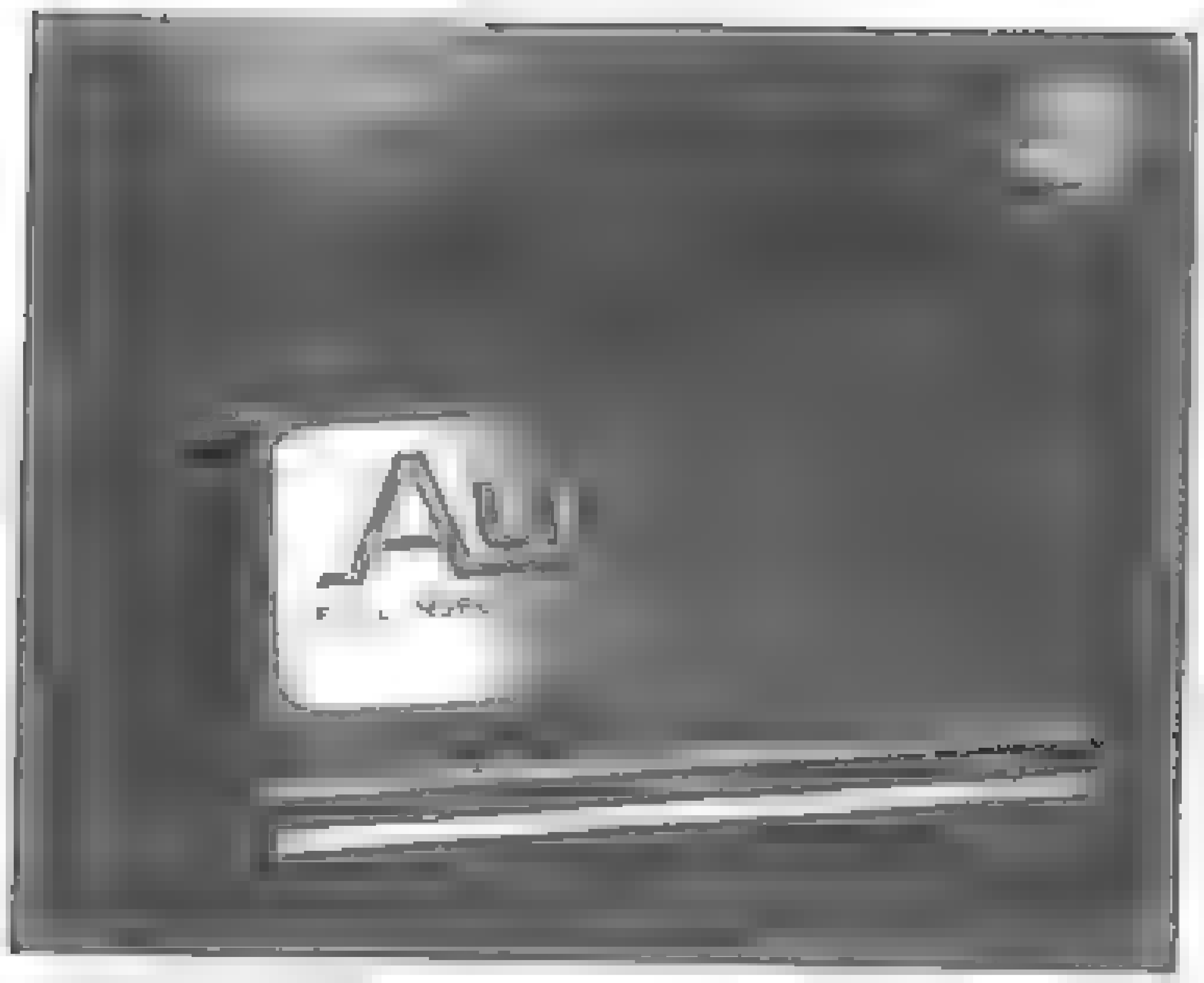
Regardless of how carefully and conscientiously you work on the dyno there is no substitute for actually driving on the road or race circuit. A great drive on the dyno seldom translates to a great drive under real-world conditions. As I have previously alluded to, you are bound to find flat spots. Pinpoint where they are occurring and fix them with minor fuel and spark tweaks. If the exhaust is popping on over run the mixture is lean on mid-range and high rpm sites at small throttle openings, so add fuel until the popping goes away. Alternatively, if the management system has provision for cutting the fuel on over-run put that feature to use and ignore any tales that you may have heard that this causes a flat spot when you get back onto the throttle. Just make sure the fuel comes back on well before the engine stalls and you won't notice any hesitation.

Before you quit mapping and pack up it is a good idea to either follow the vehicle or get someone to follow you. Look for black exhaust smoke under heavy acceleration and at steady full throttle, indicating a very rich mixture. Note that if the sun is getting low in the sky and you are looking toward it exhaust smoke appears far worse than it actually is, so if you can't see smoke with the sun behind you the fuelling is probably okay.

MAP RESOLUTION

[illegible]

For those really into this sort of thing they really should try the Autronic ECU – it has sensitivity fine enough to allow map sites as close as 1kpa (0.145psi) on the load axis and 1rpm on the other axis! However, all those sites are provided for a very good purpose. Some ECUs demand that all sites be filled in (the Haltech for example) but the Autronic simply makes the provision to enable very precise engine management during those areas of the rpm/load range where the engine suddenly gets very busy. A turbo engine may rapidly make loads of boost in unison with the variable cam



The very fine resolution possible on the Autronic engine management system mak... much easier to map on very temperamental high output rotary and turbo engines. A... he labelling is misleading, it is not a f..el only unit.

...hanism performing a big cam swing. With a ro... ...e porting and exhaus... can fully sync in an instant. Consequently ... rpm range the engin... may require major fuelling and spark ad... which may not b... ...ly met by sites at say, 3,500rpm and 4,000rpm... n fact ... ...g sites at those two points because the b... between 3,680 and 3,930rpm. Therefore it may be more effective to have a three-s... cluster in that rpm span, with the closest site either side perhaps 600–800rpm aw...

Chapter 9

The Exhaust System

In the case of the exhaust system, appearances can be misleading. It may be visually [illegible] hot gas from the cylinders. Informed modifiers, however, realise full well the importance of tuning the exhaust plumbing to improve performance. As with all other areas of the high performance engine, the exhaust system cannot be regarded as an individual entity, it is influenced by, and in turn affects, other areas, so it must be considered as a part of the whole. The aim is that the cylinders are completely scavenged of exhaust gas. On all out racing engines the exhaust system is tuned so that the exhaust gas momentum and pressure waves actually 'suck' the intake charge into the cylinder; in this way the cylinder can actually be overfilled, ie a volumetric efficiency of 101–105%. Camshaft valve overlap and the induction system both have a part to play to make this possible.

INERTIA TUNING

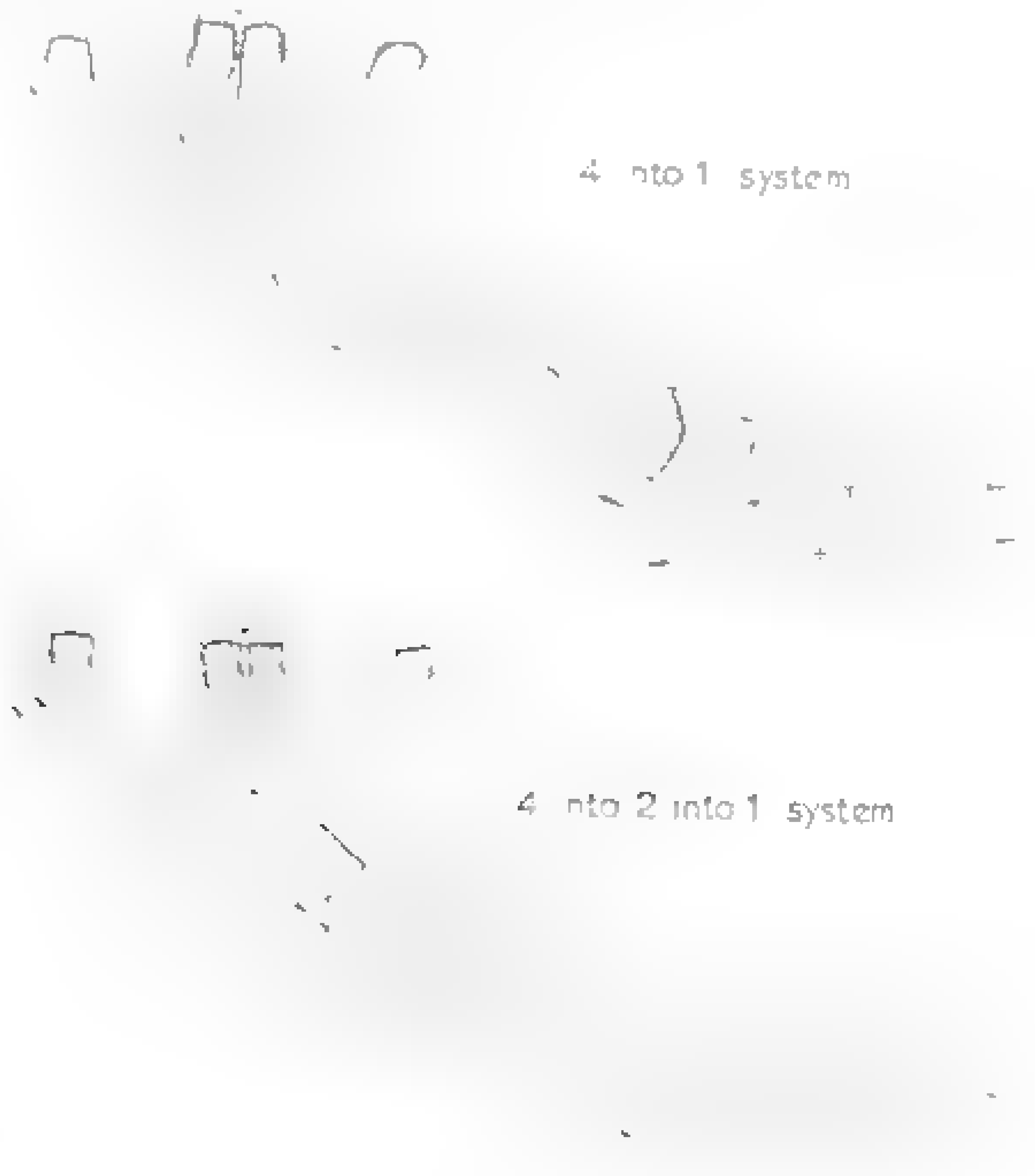
Before we discuss the 'black art' of pulse tuning we will look at what can be done to efficiently scavenge the cylinder of exhaust gas by gas momentum or inertia tuning

Figure 9.1 Exhaust valve overlap between cylinders 1 and 3

The principle of inertia tuning is that exhaust gases have weight, so once we get the gas 'rolling' it will continue to flow even after the exhaust valve has closed. This creates a partial vacuum with a resultant suction action that we can use to scavenge the cylinder. As engine speeds increase, the time available for effective cylinder exhausting will decrease, hence the need to use this suction action to empty the cylinder of exhaust gas more quickly.

Obviously if we have a gas pressure of 20psi in the exhaust manifold when the exhaust valve opens, this will restrict gas flow out of that cylinder. On the other hand, if manifold pressure is −5 to 0psi the flow restriction will be much less. For this reason we use the extractor type header with individual pipes, rather than a common manifold. The basic idea is to arrange the pipes so that the exhaust gas of one cylinder will not pressurise another. For example, look at what occurs in a four-cylinder engine where all cylinders share a common manifold (Figure 9.1). The firing order we will assume to be 1–3–4–2. At the end of its exhaust stroke No 2 tends to be pressurised by No 1; No 4 will pressurise No 3 and so on. For this reason modified cams are a waste of time if this type of manifold is retained.

Figure 9.2a Four-cylinder exhaust headers



For many years racing engines used individual pipes for each cylinder, but it is now a well-established fact that in most instances it is more beneficial to join these individual pipes together, using a collector to which we attach either a straight tailpipe or a megaphone. This arrangement picks up power and, as an added bonus, improves the power range. Figures 9.2a–d show the various header designs for four-, six- and eight-cylinder engines. The six- and eight-cylinder exhaust would usually be split into two separate systems, while the four-cylinder engine, whether in-line, vee or flat, works best with the individual pipes collected into one.

Looking at the drawings you will note there are two basic header designs for four-cylinder engines. The system giving the best power is the '4 into 1' arrangement, where the four primary pipes collect into one tailpipe. However, there are disadvantages: this type weighs more and there are usually clearance problems when trying to fit four exhaust tubes between the side of the motor and the steering gear. Another factor to be considered, particularly where four-cylinder engines are involved, is the effect of the 4 into 1 system restricting the power band. If good mid-range power is required, the '4 into 2 into 1' system is the one to adopt, although maximum power can be down by as much as 5–7% in comparison to that obtainable when using a 4 into 1 system.

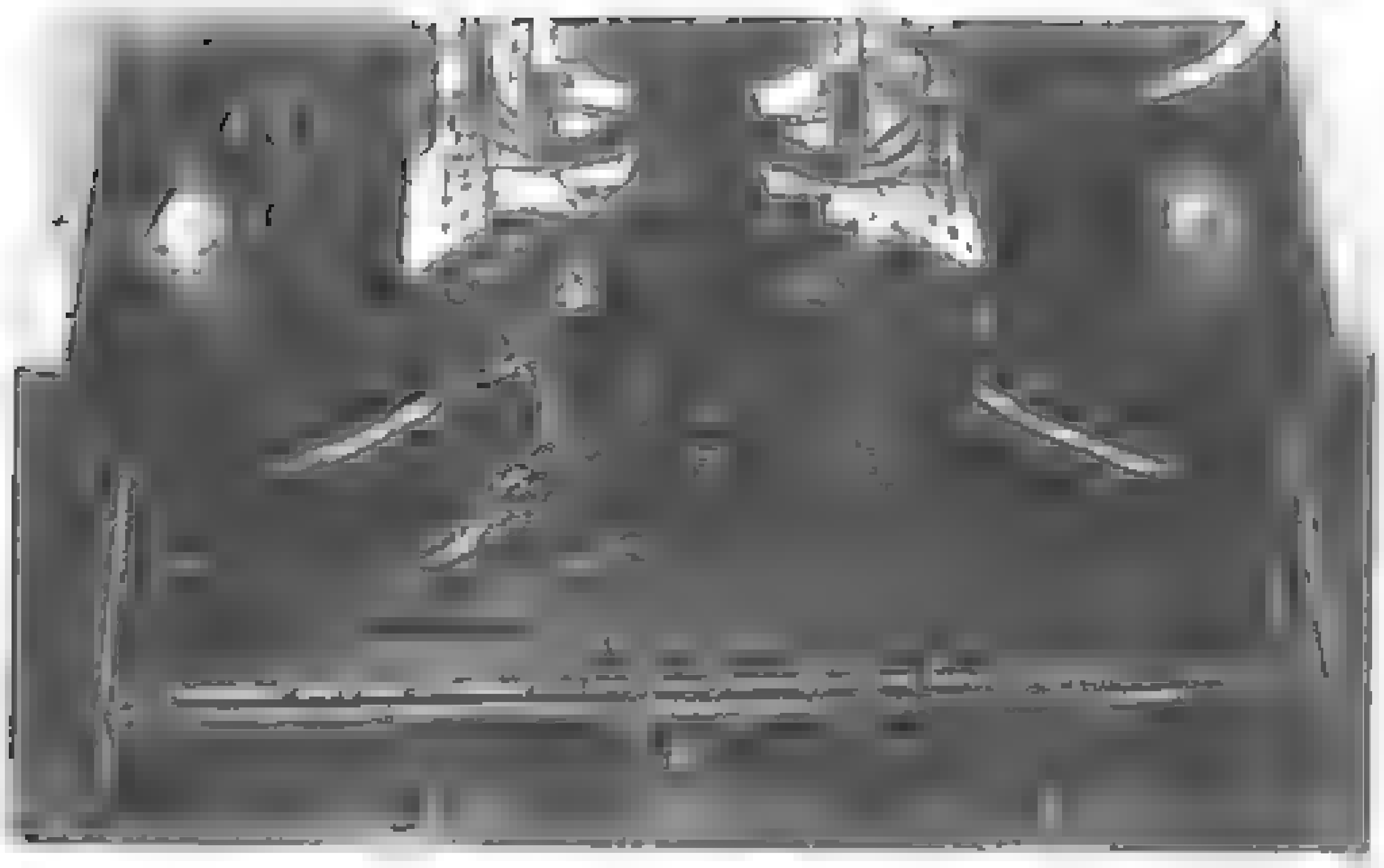
The other two four-cylinder designs (Figure 9.2b) are for Austin type engines with a siamesed centre port. The header with the long centre branch is the one to use, as the increase in volume tends to keep the pulse frequency of the centre pipe in tune with the other two branches. The second design is really only suitable if a mild cam is used.

Figure 9.2b Austin four-cylinder headers

Figure 9.2c Six-cylinder exhaust header

V8 engines pose a problem in that most use a 90° two-plane crankshaft. 180° cross-over headers will give the best power, but how you fit them under the hood of [illegible] cranks is generally used. However, as for the four cylinder header, with a few exceptions, this design isn't the best solution for most applications. If you are operating in a class where you are constantly battling a lack of traction, and where

Figure 9.2d V8 exhaust headers



F5000 engines used 4 ... n of the prim...

outright power is all important, then opt for the crossover or the 4 into 1 style a... will reduce mid range hp and sometimes give a bit more at the top-end. For a... situations choose the less popular 4 into 2 into 1 tri-Y type which for man... treated as inferior, or at best as being only suitable for street use. For tuners who ... ...ed this design it was quite a surprise to see them being utilised on Form... 5000 open wheelers run...g with a 9,000rpm limit. And it probably came ... greater surprise to see them on the big NASCAR se... ...ing at times in exc... 7,500rpm, in a class where squeezing out the maximum outright power ... ...portant. V8 engines with a two-plane crank require a pipe connection order like ... n Figure 9.

Something not to be overlooked with V8s is that the exhaust pipes from the ... should be ... Alongside the ... ...mply impossible to get the two pipes running side by side so as to be ... then you should use a balance pipe to connect between them. This should be of th... diameter as the main exhaust pipes. Unfortunately a balance pipe doe... thing like the power increase and it usually only works in the lower ba... ...he other hand, a 10–12 ... often give don... possible with a balance pipe ... the entire power band. Additionally t... ...pproach will lower exhaust noise even more th...

ACOUSTICAL TUNING

This is where pulse tuning or acoustical tuning enters the scene. The exhaust gas is expelled from the cylinder at a velocity of between 200 and 300ft per second, but pulses or pressure waves are moving through that gas at around 1,500 to 1,700ft per second. By understanding the behaviour of these waves, we can use them to improve cylinder scavenging and to increase cylinder filling with fuel/air mixture.

As the initial charge of burnt gas bursts from the cylinder into the exhaust [illegible] the gas to the end of the pipe. As it surges into the atmosphere, the positive wave dissipates and produces a negative pressure wave (suction wave), which returns along [illegible] because its pressure is much lower than the cylinder pressure. The art of exhaust tuning is to determine the length and size of the exhaust pipe for this suction wave to arrive back at the cylinder during the valve overlap period.

DETERMINING HEADER DIMENSIONS

When cam timing and valve overlap periods increase so the need to be more specific with exhaust header dimensions becomes a necessity. Part of the reason for this is that at certain rpm the cam will be causing a number of undesirable things to happen, and we can use good exhaust design to reverse, or at least minimise those problems. In addition to that, we can call on the exhaust to work in harmony with the cam, [illegible] from the cylinders and their being refilled with high density air fuel charge in the very short time frame open to us.

This means not only do we have to connect the exhaust side of the cylinders in the correct order, but we also have to select exhaust tubes of the appropriate diameter [illegible] dyno. However it is possible to achieve reasonable results if we understand the principles of exhaust tuning and then make choices in accord with those basic rules.

Header pipe diameter fixes the velocity of exhaust gas through the pipe. A large diameter pipe, relative to the size of the cylinder, will reduce gas velocity. Because engines typically produce peak torque at a mean gas velocity of around 250ft/sec the header pipe diameter in turn influences at what rpm peak torque will occur. Thus a large pipe size moves the rpm at which peak torque is produced further up the rev range.

Changing the length of the pipes tends to 'rock' the power curve of the engine around the point of maximum torque. Adding length to the pipes will increase low speed and mid-range power, with a reduction in power at maximum rpm. Shorter pipes give an increase in high speed power, at the expense of a reduction in the mid range. However there will be little change in the peak torque number or the engine speed at which it occurs (Figure 9.3).

With these factors in mind a good starting point, whether selecting an off-the-shelf header or you set out to design your own, can be found by referring to Table 9.1. For a 4-cylinder road engine or a competition engine operating at less than 9,000rpm I would always choose a 4 into 2 into 1 interference header style. Keep in mind that the length of the secondary pipe also includes the length of the first collector. All street engines regardless of the number of cylinders or the type of header will use the [illegible]

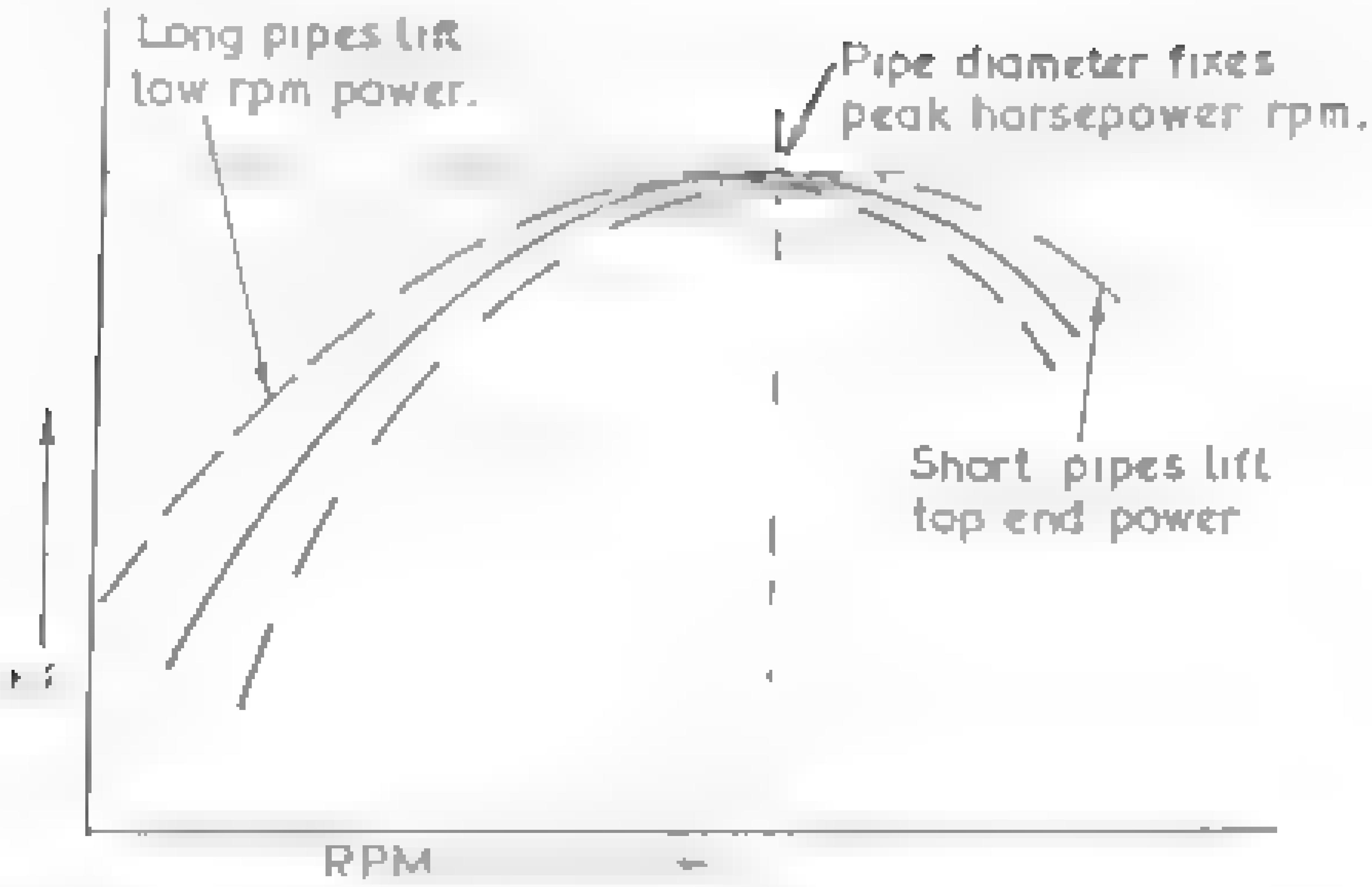


Figure 9.3 This graph illustrates how header tube diameter and length influence the shape of the power curve

smallest pipe diameters listed. Engines used for hillclimb or ski boat events would generally use an intermediate pipe diameter. In most instances, only high rpm, maximum output engines would benefit from headers using tubes anything larger than an intermediate size.

Table 9.1 Header pipe sizes for engines with competition cams

Cylinder Size (cc)	4–2–1 interference header		4–1 header 3–1 header
	Primary pipe (in)	*Secondary pipe (in)*	*Primary pipe (in)*
250	1.375 to 1.5 dia x 20 to 30 long	1.625 to 1.75 dia x 8 to 18 long	1.625 to 2.0 dia x 29 to 38 long
[illegible]	1.5 to 1.625 dia	1.75 to 1.875 dia	1.625 to 2.0 dia
400	1.625 to 1.75 dia	1.875 to 2.125 dia	1.75 to 2.125 dia
500	1.75 to 2.125 dia	2.0 to 2.25 dia	1.875 to 2.125 dia
600	1.875 to 2.125 dia	2.0 to 2.5 dia	*1.75 to 2.0 dia
700	1.875 to 2.125 dia	2.0 to 2.5 dia	*1.875 to 2.125 dia
800	1.875 to 2.25 dia	2.0 to 2.5 dia	1.875 to 2.25 dia
900	2.0 to 2.375 dia	2.25 to 2.5 dia	2.0 to 2.375 dia
1000	2.0 to 2.375 dia	2.25 to 2.75 dia	2.125 to 2.5 dia

* 4-cylinder engines would use pipe sizes 0.125 to 0.25in larger

Note all pipe lengths are as shown for 250cc cylinder and include the length from the exhaust valve to the end of the exhaust port.
The secondary pipe length also includes the length of the first collector.
All pipe diameters are outside diameters.

As mentioned previously, engines with street cams rely more on exhaust flow characteristics rather than pulse tuning. Therefore pipe diameters which provide [illegible]

9.2 this leaves header choice quite open. However for street engines I would tend [illegible]

small relative to vehicle weight

Table 9.2 Header pipe sizes for engines with street cams

Cylinder size (cc)	4–2–1 interference header		4–1 header 3–1 header
	Primary pipe (in)	*Secondary pipe (in)*	*Primary pipe (in)*
[illegible]	1.25 to 1.375 dia x 12 to 18 long	1.5 to 1.625 dia x 15 to 24 long	1.25 to 1.5 x 15 to 30
[illegible]	1.375 to 1.5 dia	1.625 to 1.75 dia	[illegible]
[illegible]	1.5 to 1.625 dia	1.75 to 1.875 dia	1.5 to 1.625 dia
500	1.625 to 1.75 dia	1.875 to 2.0 dia	1.625 to 1.75 dia
600	1.75 to 1.875 dia	1.875 to 2.125 dia	+*1.625 to 1.75 dia
700	1.875 to 2.0 dia	2.0 to 2.125 dia	*1.75 to 1.875 dia
800	1.875 to 2.0 dia	2.0 to 2.125 dia	1.75 to 1.875 dia
900	1.875 to 2.125 dia	2.0 to 2.25 dia	1.875 to 2.0 dia
1000	1.875 to 2.125 dia	2.0 to 2.25 dia	1.875 to 2.0 dia

*4-cylinder engines would use pipe sizes 0.125 larger
+ V6 engines would use pipe sizes 0.125 larger

Note. all pipe lengths are as shown for 250cc cylinder
The secondary pipe length also includes the length of the first collector
All pipe diameters are outside diameters

Rather than simply choosing a header from a simple table I prefer to do some [illegible]

wide variety of applications

The formula to work out the primary pipe length is:

$$P = \frac{850 \times ED}{rpm} - 3$$

where rpm = the engine speed to which the exhaust is being tuned, and ED = 180° plus the number of degrees the exhaust valve opens before BDC.

To make the task simpler I have prepared Table 9.3, so that the primary length can be read straight off. Generally, road motors will require a manifold tuned to work at maximum torque rpm. Racing motors, on the other hand, use a header tuned to work at either maximum horsepower rpm or at a speed midway between maximum torque and maximum hp revs.

Table 9.3 Exhaust primary pipe length (in)

rpm	Exhaust valve opening degrees before BDC								
	50°	*55°*	*60°*	*65°*	*70°*	*75°*	*80°*	*85°*	*90°*
4,000	46.0	46.9	48.0	49.0	50.2	51.2	52.1	53.3	54.4
4,500	40.5	41.3	42.3	43.2	44.3	45.1	46.1	47.1	48.1
5,000	36.2	36.9	37.7	38.6	39.5	40.4	41.1	42.0	43.0
5,500	32.6	33.2	34.1	34.8	35.7	36.4	37.1	38.0	38.9
6,000	29.6	30.3	31.9	31.7	32.5	33.1	33.8	34.5	35.4
6,500	27.2	27.7	28.4	29.0	29.8	30.4	31.0	31.7	32.4
7,000	25.0	25.5	26.1	26.7	27.4	28.0	28.6	29.2	29.8
7,500	23.1	23.6	24.2	24.7	25.4	25.9	26.5	27.0	27.6
8,000	21.5	21.9	22.5	23.0	23.6	24.1	24.6	25.2	25.8
8,500	20.0	20.5	21.0	21.5	22.1	22.5	23.0	23.5	24.0
9,000	18.7	19.2	19.6	20.1	20.6	21.1	21.5	22.0	22.5
9,500	17.6	18.0	18.4	18.9	19.4	19.8	20.2	20.7	21.2
10,000	16.5	16.9	17.4	17.8	18.3	18.6	19.1	19.6	20.0
10,500	15.6	16.0	16.4	16.8	17.2	17.6	18.0	18.4	18.8
11,000	14.8	15.1	15.5	15.9	16.3	16.7	17.0	17.4	17.8
11,500	14.0	14.3	14.7	15.1	15.5	15.8	16.2	16.6	17.0
12,000	13.3	13.6	14.0	14.3	14.7	15.0	15.4	15.8	16.1

Once the primary pipe length has been determined, we can then work out the inside diameter using the following formula.

$$ID = \sqrt{\frac{cc}{(P + 3) \times 25}} \times 2.1$$

where cc = cylinder volume in cc, and P = primary length in inches

Headers for road engines usually work well enough if the pipes are of the same diameter as the exhaust valve, but racing engines demand more exactness than this if we are to achieve ultimate performance. In using the above formula, sizes will have to be worked to suit exhaust tubes that are available commercially

If a '4 into 2 into 1' system is preferred, we use the same formula or Table 9.3 to work out the total length (P) of the header pipes, which will be the combined length of the primary pipe (P1) plus the length of the secondary pipe (P2). The inside diameter can then be determined for the four primary pipes (P1) using the same formula

$$ID = \sqrt{\frac{cc}{(P + 3) \times 25}}$$

Once the inside diameter of the primary pipes is calculated, we can then work out the inside diameter of the two secondary pipes (P2) by the formula

$$IDS = \sqrt{ID^2 \times 2} \times 0.93$$

where ID = the calculated inside diameter of the primary pipes (P1)

This 5ltr V8 Supercar engine has very short secondary pipes which helps top-end hp. [illegible] ronger bottom end. Note that as [illegible]

The length of the primary pipes (P1) should always be at least 15in. Adding length t[illegible] the primaries will favour top-end power, while adding it to the secondaries helps th[illegible] [illegible]. The length of the secondary pipes (P2) can be found by simple subtraction [illegible]

[illegible] calculations for the '4 into 2 into 1' system [illegible] used to determine the [illegible] r the long centre [illegible] ch type Austin header: the only difference is that th[illegible] [illegible] is the [illegible]e diameter as the second[illegible] (IDS) for the full length (E[illegible]

PRACTICAL HEADER TUNING

In theory it looks very simple to arrive at a header design with pipes of pre[illegible] [illegible] length and diameter, unfortunately in practice it does not often work [illegible] [illegible]. A header constructed to the formulae outlined will work reasonably wel[illegible] [illegible] basis for further experimentation on the dyno or at the race t[illegible] [illegible] to variables in the design of cams, inlet manifolds, cylinder h[illegible] [illegible] length and diameter of the header pipes will have to be changed abo[illegible] [illegible] eal size for your engin[illegible]

[illegible] find that the engine's torque peak is at 7,000rpm and you want maximu[illegible] [illegible] 6,000rpm, reduce the pipe diameter. Generally, a reduction i[illegible] primary [illegible] [illegible] 0.125in will move the torque peak down by 500-600rpm [illegible]

the diameter of the headers will raise the engine speed at which maximum torque occurs by approximately the same rpm for each 0.125in increase.

Table 9.4 shows the results of testing headers of various sizes on an Opel 2-litre race engine. As can be clearly seen, the 2⅛in headers really only worked very high up in the rpm range, but as this engine would be competing in a class with an enforced 8,500rpm limit, the power loss in the lower ranges would not be compensated for by the modest power increase seen above 7,750rpm. The 2in pipes provided a much superior power spread while losing out to the 1⅞in headers below 7,000rpm.

Table 9.4 2-litre XE Opel header comparison

	Test 1		Test 2		Test 3	
rpm	hp	Torque	hp	Torque	hp	Torque
5,250	164.8	164.9			169.9	170.0
5,500	172.5	164.7			182.5	174.3
6,000	204.0	178.6	196.7	172.2	202.8	177.5
6,500	221.3	178.8	212.5	171.7	224.3	181.2
6,750	231.5	180.1	225.7	175.6	234.9	182.8
7,000	243.1	182.4	235.1	176.4	241.1	180.9
7,250	249.3	180.6	245.2	177.6	245.8	178.1
7,500	253.9	177.8	255.8	179.1	253.8	177.7
7,750	256.5	173.8	258.5	175.2	259.3	175.7
8,000	264.9	173.9	267.2	175.4	265.7	176.4
8,250	275.2	175.2	276.2	175.8	270.3	172.1
8,500	273.0	168.7	276.9	171.1	264.9	163.7
8,750	268.7	161.3	273.2	164.0	260.4	156.3
9,000	264.2	154.2	269.9	157.5		

Test 1 – 2in race headers, 4 into 1

Test 2 – 2⅛in race headers, 4 into 1

Test 3 – 1⅞in race headers, 4 into 1

Note: in all tests the same primary pipe length was used and the tailpipe length and diameter was not changed.

Previously I stated that changing the length of the pipes tends to 'rock' the power curve of the engine around the point of maximum torque. Adding length to the primary pipes will increase low-speed and mid range power, with a corresponding reduction in power at maximum rpm. Some race engines, however, do not conform to this general rule and will show a good increase in mid-range power with very little loss at the top end even with substantial increases in primary pipe length. Usually we expect to see an engine respond in a similar way to a 4in increase in primary pipe length as it would to 0.125in decrease in pipe diameter. Thus a 350 Chev race engine would work pretty much the same with 1⅞in primaries 34in long as with 1¾in pipes 30in long. However, the V6 Buick (and some others) keeps gaining bottom-end power without knocking the top end with primaries lengthened to around 44in, about 12in longer than comparable competition engines.

Unfortunately many racers only concern themselves with developing an exhaust that gives the best power over the largest rpm range without ever understanding that

they can use a number of different headers to alter the power characteristics of the engine to better suit different tracks and surface conditions, etc. When traction is not a problem, lots of mid-range grunt will help drive the car out of corners more rapidly. However, that power, which was so desirable at one circuit, may actually slow the car or overwhelm the driver at another circuit, or under different track conditions. A relatively quick fix for a problem like this is to swap the headers to a set that takes away a rush of power in the mid-range. Note, however, that engines utilising computerised ignition and fuel injection may require a 'chip' change when the header is changed.

The dirt speedway is a typical example of a track with quickly changing surface conditions. Early in the night the track might be very heavy, calling for a header that [illegible] means a header with 1⅝in primaries 30in long going into a 3in tailpipe. After a few hours of racing the track may dry out and the surface could harden like concrete. Such conditions may call for a shift to more top-end power. A suitable header may have 'stepped' primaries 1¾in for the first 13in, stepping up to 1⅞in for 15in, terminating in a 3½in tailpipe. On the other hand, the track surface may break up badly as racing progresses, so to kill bottom-end power headers with 2in tubes 30in long leading into a 3in tailpipe may be fitted.

A similar situation exists in club-type rallying. Some events may be smooth, dry tarmac. Here an Escort with a 1700 BDA Cosworth may benefit from a stepped header that starts out at 1¾in for 10in, then steps up to 1⅞in for 18in, then into a 2½in tailpipe so as to lift power in the upper range. However, such power characteristics could prove to be a handful to a novice when that same tarmac is snow covered. Under such conditions we want the engine to respond very gently to delicate throttle movements – we do not want sudden surges of power. The approach here may be to use unequal-length stepped headers.

UNEQUAL LENGTH PIPES

Why unequal length primary pipes? Well, this tunes the torque peak in each cylinder to a different rpm, thus flattening and smoothing the torque curve. It tunes out that [illegible] torque peak will be lowered slightly but may go all the way from, say, 5,500 to 6,500rpm. Hence one cylinder may peak at 6,000rpm, another at 5,750, another at 5,500 and still another at 6,250.

To achieve this the primary length would be adjusted to minus 2in plus 4in from the ideal length. Therefore if the 'correct' length was found to be 29in, the four pipes would vary in length from 27in to 33in. In reality I prefer unequal length headers in most applications. For a long time it was thought that equal-length tuned headers were the best way to go, but when it was discovered in installations where equal-length pipes just would not fit that the package often became more drivable, some tuners slowly came around to acknowledge the advantages of unequal-length pipes.

In restricted classes equal-length primaries are more important, as here the chassis, tyres or driver are less likely to be overwhelmed by an excess of power. In these categories of competition the breathing may be impeded by a single small carburettor, or power may be pegged by a low compression ratio of 9:1 and perhaps the use of lower-octane pump fuel rather than race fuel. Such imposts rule out a sudden

rush of power, so every cylinder must be tuned to make maximum hp at every point in the usable power band. This calls for carefully designed equal-length headers.

STEPPED HEADERS

Earlier mention was made of stepped headers, ie primary pipes with 'steps' or changing pipe diameters. Those who follow any form of two-stroke competition know that the two-stroke exhaust is formed from a series of cones. The first part, also known as the header pipe, usually tapers at somewhere between 1.25 and 2.2°. This is because in terms of physics a cone provides the best exhaust flow, and when the angle is shallow it preserves gas momentum while keeping flow restriction and turbulence to a minimum. Obviously a stepped header more closely approximates this ideal than a tube of constant diameter.

A typical stepped header would have just one or, in a few cases, two steps, with the header diameter increasing 0.125in at each step. At times engines that like very long primaries work best with a quarter-inch step. An example is the V6 Buick, which usually works well on road courses with primary tubes of 1⁷/₈in diameter for the first 11 or 12in, stepping up to 2¹/₈in tubes 34in long.

Table 9.5 2-litre XE Opel stepped header comparison

	Test 1		Test 2	
rpm	hp	Torque	hp	Torque
5,500	172.5	164.7	179.5	171.4
6,000	204.0	178.6	206.5	180.8
6,500	221.3	178.8	224.9	181.7
6,750	231.5	180.1	234.3	182.3
7,000	243.1	182.4	243.5	182.7
7,250	249.3	180.6	251.4	182.1
7,500	253.9	177.8	257.3	180.2
7,750	256.5	173.8	263.8	178.8
8,000	264.9	173.9	268.7	176.4
8,250	275.2	175.2	277.4	176.6
8,500	273.0	168.7	276.3	170.7
8,750	268.7	161.3	275.2	165.2
9,000	264.2	154.2		

Test 1 – 2in race headers, 4 into 1

Test 2 – 1⁷/₈in to 2 to 2¹/₈in stepped race headers, 4 into 1

Note, in addition to being stepped, the headers in Test 2 had primary pipes 3in longer than those used in Test 1

In Test 2, phasing of both inlet and exhaust cams was altered slightly to optimise improved exhaust scavenging

Because of improved engine efficiency, the ignition curve had to be modified with about 1° less advance up to 7,250rpm and 2° less from 7,500rpm up to the rev limit in Test 2

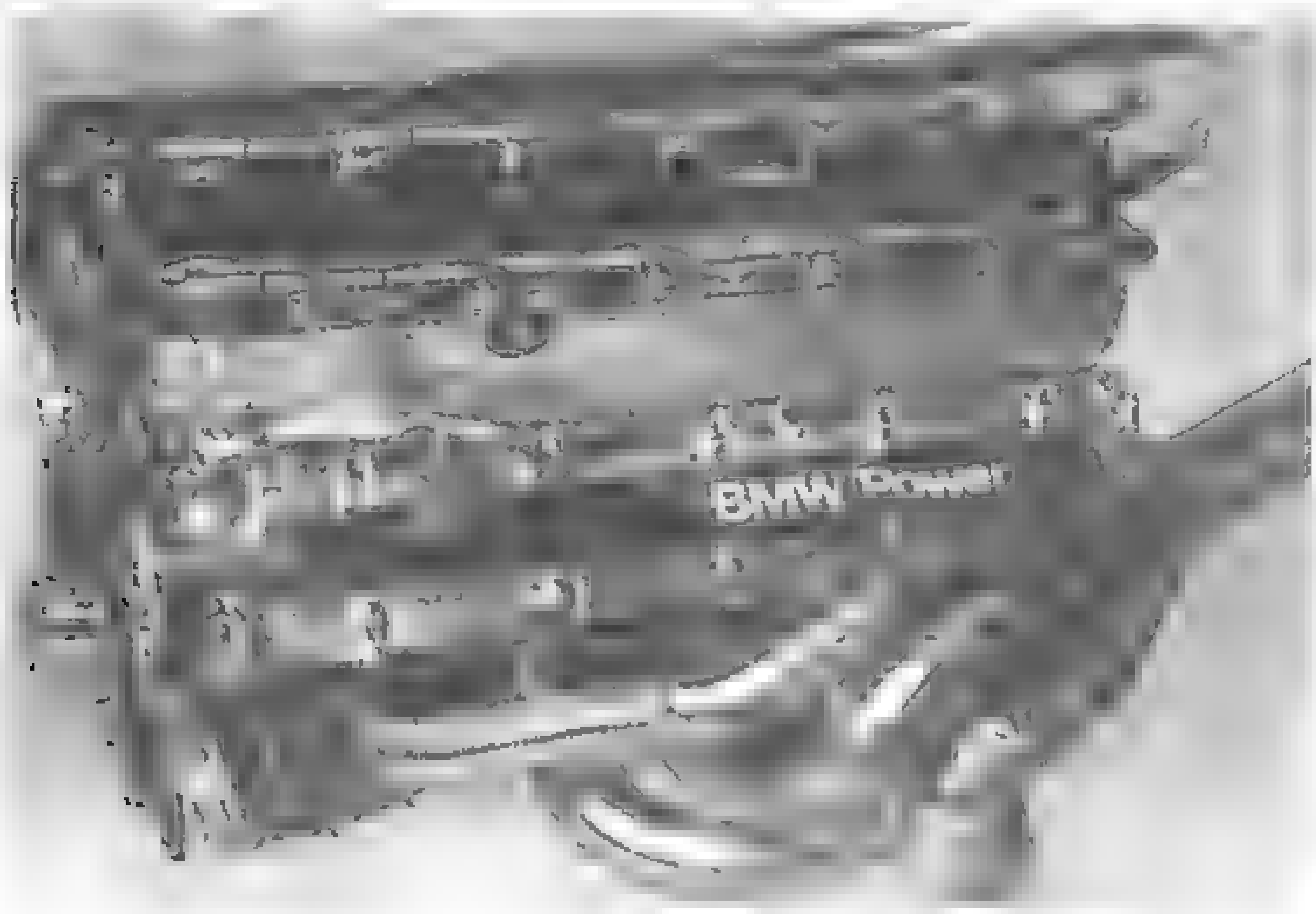
A lot of time can be spent figuring out where to step the primaries, but you can usually reckon on the first step being about 10 to 14in from the exhaust port. In theory, if the first section of pipe is relatively small to improve bottom end power, exhaust reversion will occur at around the 10 or 11in mark, so that is where the first step should go. On the other hand, if the pipe size is quite generous, favouring more the top-end of the power range, the first step will probably want to be closer to the 14in point to avoid losing bottom-end power.

Where should the second step be located? That takes even more dyno time to get right. Some say that if the primary length is 30in, the steps should be set at equal intervals, ie with the first step at 10in and the second at 20in. If only a tuner's life were so simple! To give you some guidance in this I feel it makes more sense to figure how far back from the collector the second step should be.

When the overall primary length is reasonably short, ie 24 to 28in, I would begin testing with the second step around 0.2 times the primary length back from the collector. Thus a 25in primary would have the second step marked off 5in from the collector. However, as the primary length increases to around 36–40in, the second step will be located closer to 0.25 times the primary length before the collector.

Depending on which way you step, a carefully developed step header can add around 500rpm to either the bottom or the top of the power band. Therefore if your current header has 1⅞in tubes and you want more bottom end, step down at the exhaust port to 1¾in. However, if the bottom end is already acceptable and you want more top end, you would continue to come off the head in 1⅞in tube, then step up to

Stepped headers can bring a power gain and broaden the power band of most types of engines. Originally it was thought that stepping wasn't applicable to very high rpm engines, or those with cylinders smaller than about 450cc



2in at the first step. Assuming that the bottom end is still intact after this change, you may like to try introducing a second step and go up to 2 1/8in pipe at that point.

When the 2-litre Opel mentioned earlier was re-tested with a stepped header, it produced the figures shown in Table 9.5. Clearly the engine responded very well to a header that stepped from 1 7/8 to 2 to 2 1/8in. Previous dips in the power curve have been smoothed out, and in addition there is a little more power right on the limiter and also down at 5,500 and 6,000rpm.

Don't assume that stepped headers are the way to go on all engines – they are not. To gain the theoretical benefits can involve a lot of dyno time, perhaps time and money that may reap richer rewards elsewhere. Also if your style of competition doesn't demand a broad power range stepping will most likely be a waste of time, or perhaps even a handicap if you are already struggling for traction in the middle of the power band. On the other hand, if you have a good size budget, don't be quick to conclude that stepping won't work on your engine. I started out thinking that it wasn't worth the effort and expense in engines operating at very high rpm, or with small cylinders. Now of course you would be hard pressed to find factory racers from the bike manufacturers that don't have tapered header tubes, or Formula 1 engines that don't run stepped headers.

THE COLLECTOR'S INFLUENCE

The next part of the header, the collector, is where the individual primary tubes come together as one. Again this is an area where the serious tuner has the option of choosing from a number of collector styles to 'tune' the characteristics of the header. Remember that, regardless of the style chosen, the collector can only boost torque at engine rpm below the torque peak, the shape of the power curve above the torque peak being determined by the primary pipe diameter and length.

Generally we want to boost engine torque whenever possible. However, as I've previously stated, in some circumstances excessive torque at lower rpm may be a hindrance, actually reducing overall performance. Consequently in some classes of drag racing the pipes from individual cylinders are not brought together with a collector as any increase in mid range torque just makes traction problems more difficult. You may have witnessed the same sort of thing with sprint cars on dirt speedways. When there are no noise rules and when traction is limited you will see these cars racing without collectors so as to get rid of unwanted torque.

Going back to the introduction to this chapter, you will recall that exhaust theory revolves around using exhaust gas momentum or inertia and pressure waves to scavenge exhaust gas from the cylinder in the first instance, then, during the valve overlap period, the waves begin to move the intake charge down the inlet tract and into the cylinder. When the exhaust valve closes any vacuum 'energy' left remaining in that particular primary pipe will be lost if that pipe is not connected to another primary pipe. Using a collector to connect the primary pipes together allows this unused, left-over vacuum to lower the pressure in all of the other primary pipes, thus assisting exhaust flow out of those cylinders that have open exhaust valves.

However, once past the torque peak rpm there is not enough time for the primary tube to 'empty' of exhaust gas after the valve closes, so it is unable to 'pull' a vacuum on any of the other primary pipes to which it is connected. Obviously if the primary

pipe was made very short, vacuum would reach down to the collector at higher rpm and draw on the other primaries, but shortening the primaries has the undesirable effect of killing bottom-end power, so this action would only work on tracks where a narrow power band was acceptable

BAFFLE COLLECTOR DESIGN

Figure 9.4 illustrates an ordinary collector that I personally prefer to call a 'baffle' [illegible] like this a pressure wave returns up the primary Now if a negative pressure wave (suction wave) arrives back at the exhaust port during the time when the exhaust valve is next open, it will work in harmony with any vacuum created by exhaust gas [illegible] rpm a positive pressure wave will arrive just as the exhaust valve is about to close to reduce the amount of fuel/air charge that 'spills over' (called over-scavenging) into the exhaust during the valve overlap period due to the effect of exhaust gas momentum

That is the positive aspect of what this type of exhaust collector will accomplish The negative side is that at certain rpm a positive pressure wave will arrive back at the exhaust port during the early phase of the exhaust valve opening period and actually impede exhaust flow out of the cylinder. Likewise, in other parts of the rev range a negative pressure wave will arrive just as the exhaust valve is closing and draw even more air fuel mixture out into the exhaust Obviously both scenarios lead to reduced hp and increased fuel consumption

Consequently this type of collector is better suited to road engines and competition engines utilising milder race cams with a duration of up to about 260° measured at 0 050in lobe lift, and lobe centres in the range of 105–110° In this kind of application the baffle collector works very well

Some competition headers using a baffle collector have a 'spear' welded to the centre of the baffle where all the primaries terminate. This is supposed to reduce flow turbulence, but I believe that devices like this are really just a marketing gimmick. The main thing to watch with any collector is the angle at which it tapers In general terms

Figure 9.4 Baffle-type collector

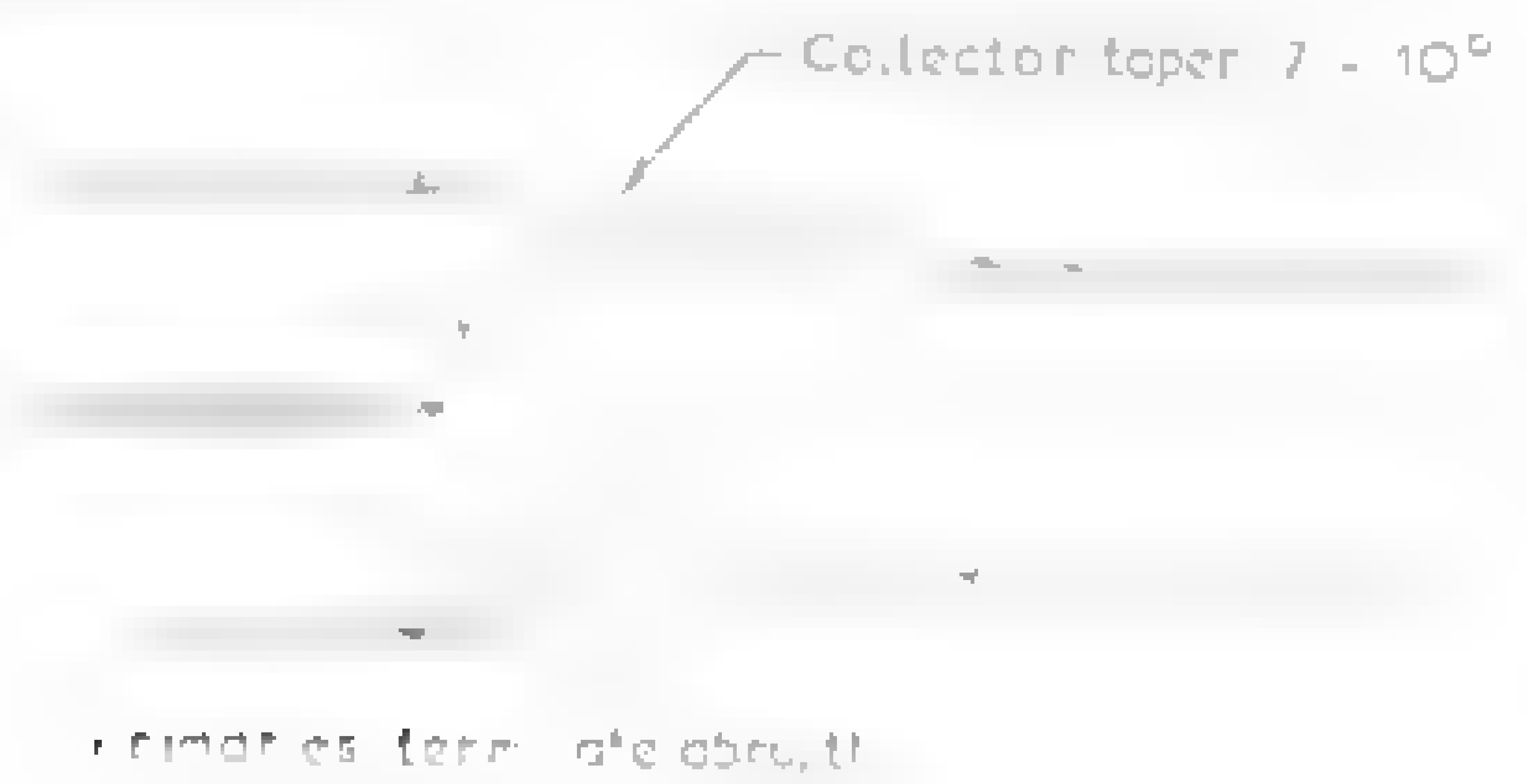


Figure 9.5 Merge-type collector

I have found the best angle to be 7 or 8° (14 or 16° included angle). However, some engines will work best with a steeper taper of 9°, or even up to 10°. Very shallow tapers of less than 7° do not seem to work too well, probably due to excessive collector volume allowing massive flow turbulence.

MERGE COLLECTOR DESIGN

The collector that is better suited to maximum-effort race engines with long-duration, big-overlap cams is known as a 'merge' collector (Figure 9.5). With this type the primary pipes do not terminate abruptly, but 'merge' or blend into the tailpipe. This has the effect of reducing the intensity of reflected pressure pulse waves in the [illegible] mentioned previously. Of course, using this type of collector means that the headers lose out on the positive aspects of pulse tuning. However, as we are talking about maximum-effort engines, operating over a very narrow rev range, this gives the [illegible] relying on exhaust gas momentum or inertia alone.

With the merge design the primaries must generally come together fairly abruptly [illegible] and a number also work best with slightly shallower tapers around 8½ to 9°.

VENTURI MERGE COLLECTOR

Another collector, which many also refer to as a merge collector, is what I prefer to call a 'venturi merge' collector. Referring to Figure 9.6 you will see that this design necks down, then flares out into a cone to meet the tailpipe. This has the effect of picking up gas speed to boost the scavenging power of the header, which serves to extend the rpm range over which inertia tuning works. Venturi merge collectors seem to work best when the tailpipe diameter is very large relative to the primary tube diameter, so I suspect forcing the gas to accelerate through a relatively small venturi further serves to diminish the amplitude of pressure waves travelling within the primaries, thus ensuring that the header is working more in concert with exhaust gas momentum and is less influenced by pressure pulse waves.

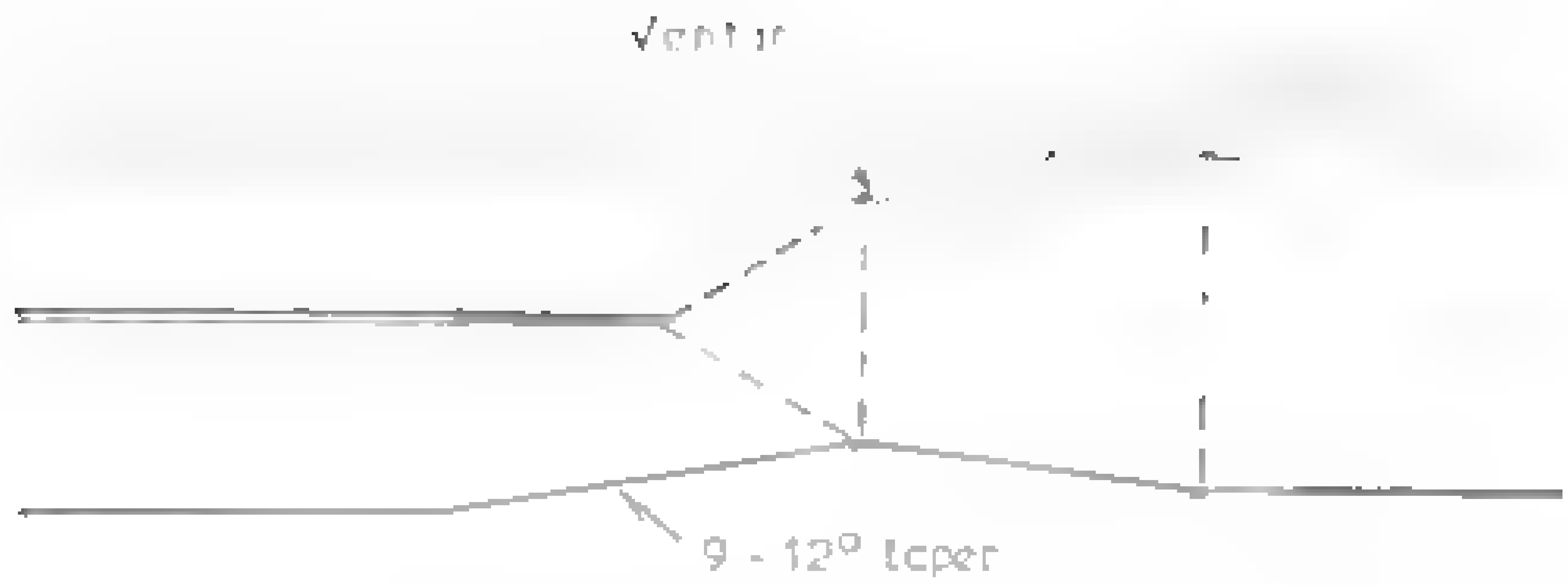


Figure 9.6 Venturi merge-type collector

Whereas a typical race header with a merge-style collector may have, say, 2in primaries exhausting into a 2½in tailpipe, a venturi merge collector would be used if a large tailpipe of around 3 or 3½in was necessary in V6 and V8 race engines with cylinder capacities up around 700 to 800cc exhausting into primaries around 1⅞in to 2⅛in. As with the ordinary merge collector, the venturi type has the primaries converging at around 10°. That is where I would begin testing, and I would then set the diffuser cone taper at 8° This may not be the ideal taper, but it will be close Anything in the range of 7–10° will work – it is just a matter of dyno time and track time to determine what works best

While these two angles are important, what is equally important is the actual diameter of the venturi A rough rule of thumb is to make the venturi half an inch

venturi becomes 2½in. The final size may end up anywhere from 2⅜in to 2¾in, but what I have suggested is a good starting point

SPLIT INTERFERENCE COLLECTOR

The fourth style of collector, which I call a 'split interference' collector, has a number of virtues, one of which is that it hides from prying eyes just what style of header you are running! Figure 9.7 illustrates how this collector connects the primaries to a 4–2–1-type configuration. However, whereas the conventional '4 into 2 into 1' header uses fairly short primaries and (usually) comparatively longer secondary tubes, this split interference style collector provides an arrangement of long primaries and short secondaries that may give a useful increase in mid range power without drastically killing off the top end. This occurs because any vacuum created in a primary tube by exhaust gas momentum will not be 'diluted' by what is taking place in the other three tubes to which it is connected Instead, with this collector only one other pipe is involved, so at lower gas speeds, when vacuum levels are fairly poor, the full influence of that vacuum will be felt by the other primary to which it is connected This is what serves to pull up mid-range power Then at high rpm, because this design does not have long secondaries to restrict exhaust flow, there is only minimal hp loss at the upper end of the power band

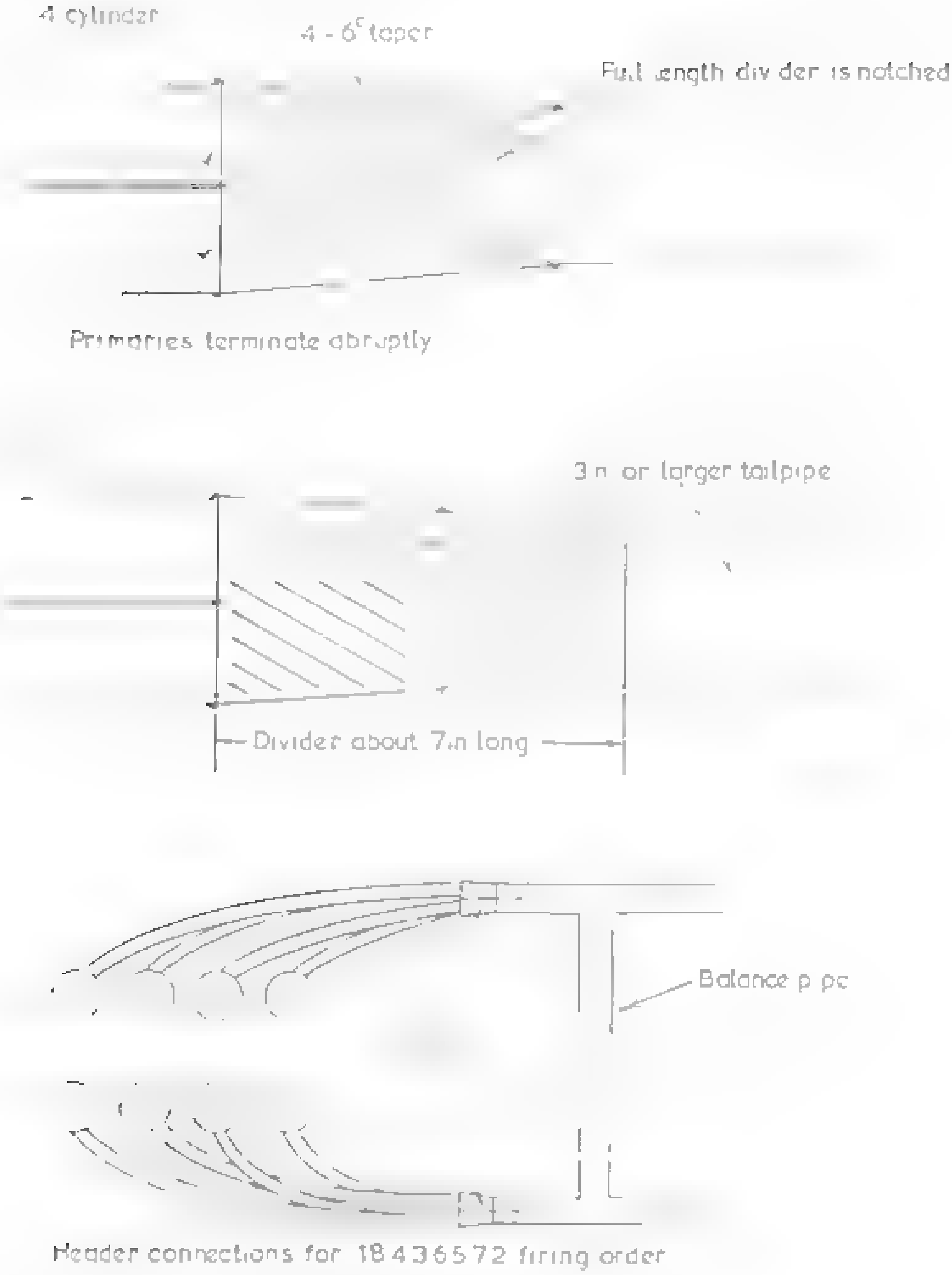


Figure 9.7 Split baffle collectors

With this collector attached to a four-cylinder engine, primary pipes of cylinders firing 360° apart are connected. However, with V8 engines primaries alternatively firing 270° and 450° apart are joined, which tends to produce a torque peak at different rpm in each cylinder, thus flattening the torque curve much like unequal-length headers. Additionally, because with this style the collector volume is halved, there is reduced flow turbulence at this point, which tends to help the torque curve at a number of places.

Some people run this type of collector at all times, but I feel that the time to resort to the split interference collector is when there is no time to change the headers to suit changes in track conditions. When you need more bottom-end power quickly it only takes a few minutes to swap over from another collector type to a split interference type. My suggestion to get the best from this is to build your headers with the primaries about 3–4in shorter than what is required. When you want a true '4 into 1' header, slip on a collector that includes primary extensions 3–4in long to give the correct tuned length. Then when you want more bottom-end grunt, it is a simple thing to remove that collector and attach a split interference type, which in effect will add something like 6–8in as a secondary to the header. However, because the primaries are 3–4in short anyway by design, the actual tuned length will increase by 2–4in, which helps the mid range without knocking the top end by very much.

A collector fashioned like this will not benefit all engines. Mid-size V8s seem to be the most inconsistent, with some responding favourably, some indifferent and a few losing out everywhere. V8s under about 5 litres mostly work well, as do monsters over 8 litres, but in between results are patchy. Four-cylinder engines, however, almost always respond nicely to the split interference collector.

When building a collector for a V8 I would begin with a divider 7in long, which means that it will extend into the tailpipe. Four-cylinder engines, which usually work best with a relatively small-diameter tailpipe, will suffer power losses if this approach is followed, so it is preferable to use a long, shallow collector with a 4½ to 5½° taper. The divider can run the full length of the collector, but it must be notched so that gas flow into the tailpipe is not unduly restricted. A short notch increases the venturi effect and serves to broaden the power band, while a longer notch reduces this effect but improves higher rpm gas flow.

Using the following formula we can work out the length of the collector:

$$\text{Collector length} = \frac{ID2 - ID3}{} \times \text{Cot } A$$

= diameter of the collector outlet, and

12.7, 5½° = 10.4, 7° = 8.15, 8½° = 6.7.

TAILPIPE DIMENSIONS

When it comes to the tailpipe there are a couple of ways in which we can establish a baseline size to begin dyno testing. When an open unsilenced exhaust is used, the combined length of the collector and tailpipe will be the same as the length of the ... as

primary pipe plus 3in. Then the tailpipe inside diameter is calculated using the following formula

$$ID3 = \sqrt{\frac{cc}{(P + 3) \times 25}} \times 2$$

where ID3 = tailpipe id in inches, cc = volume of one cylinder in cc, and P = primary length in inches

The alternative is to set the combined collector and tailpipe length the same as the primary length, then from Table 9.6 read off the tailpipe outside diameter based on engine hp output

Table 9.6 Tailpipe size

hp	pipe od (in)
80–120	1⅞
110–140	2
130–150	2⅛
140–185	2¼
180–220	2½
210–265	2¾
250–320	3
280–360	3½
400–500	4
480–630	4½
580–750	5

Note when two exhausts are used divide engine hp by two

Establishing a suitable tailpipe size is becoming more complex with road cars because of the addition of the catalytic converter to reduce tailpipe emissions. Because CATs typically add considerable heat to the exhaust gas, it is not unusual to find that an increase in exhaust diameter of 0.125in to 0.375in over the size indicated in Table 9.6 from the CAT back, will show some power increase

CUTTING DOWN ON NOISE

Likewise more and more classes of competition now make the use of CATs mandatory, and noise limits have caused tuners to consider mufflers and a number of other measures to avoid pushing noise meters past the regulated level, leading to possible disqualification

Conventional thinking among some is to build a maximum-hp unsilenced exhaust, then add to it what is required to meet noise regulations and accept the inevitable large power losses. Approached from this level of thinking it is obvious that a lot of hp will be knocked off, perhaps up around 10%. Some shrug this off, saying that everyone else has to meet the same noise limits so that

amount. The truth is that some will not be losing even 5% because they have invested dyno and track time to see what works and what does not.

The very first requirement is to read carefully the regulations and see what the rules are; if the rules do not say that you cannot do something, then assume that you can legally. Hence if the rules do not state that the tailpipe must terminate at the rear of the car, do not run it out the back unless doing so helps you gain hp or lower noise. Likewise, do not run the exhaust out of the side where it points at the noise gun just because everyone else does; the idea is to aim the tailpipe away from the sound meter. So if you run anti-clockwise on speedways and the noise meter is on the infield, aim the tailpipe out of the right side with a turn-down tip, so the maximum amount of sound is absorbed by the track rather than being reflected back to the meter by concrete walls or empty seating shelters. On road courses, where meter locations vary from track to track, it is usually possible on sedan cars to design the exhaust so that it can be swapped to exit on either the left or the right.

To avoid using a muffler, or to permit a small, low-restriction resonator to pass noise tests, some tuners have tried all sorts of exhaust tricks. Their reasoning is, keep away from mufflers at all costs, they just kill power. Therefore even when the rules do not call for a rear-exit exhaust, they will run a full length pipe, and perhaps step it down a couple of sizes hoping to get the noise down. The thinking is that since there is no measurable back pressure in the system, there cannot be any loss of power. Well, let me assure you that I have seen 2½% losses in both hp and torque in systems like this. The relationship between back pressure and hp is not at all clear at the lower end of the pressure scale.

Table 9.7 details the results of tests on a 223hp race engine with various exhaust and muffler combinations. Clearly there is a fairly fuzzy relationship between back pressure and power, and for that matter between noise and power. Also note that both purpose-made race mufflers, which I have numbered 1 and 7, failed the noise tests, and

Table 9.7 Exhaust and muffler tests

Description	hp	% loss	back pressure	comments
open tuned tailpipe	223.2		0	tailpipe 26in x 2½in
side exit extension (A)	221.8	0.6	0	plus 42in extension
rear exit exhaust (B)	217.3	2.6	0	2½in tailpipe, 129in overall
big-bore rear exit (C)	220.4	1.2	0	3in system
exhaust (C) fitted with				
muffler 1	220.2	1.3	0	failed noise test
muffler 2	216.0	3.3	2.04	passed noise test
muffler 3	215.5	3.5	0.4	failed noise test
muffler 4	216.7	2.9	1.1	easily passed noise test
muffler 5	217.3	2.7	1.37	easily passed noise test
muffler 6	218.4	2.2	1.45	just failed noise test
muffler 7	210.4	5.8	1.2	failed noise test
exhaust (A) fitted with				
muffler 4	218.4	2.1	0.9	easily passed noise test
muffler 5	218.8	2.0	1.23	passed noise test
muffler 6	219.7	1.6	1.3	just passed noise test with exit away from sound gun

In addition muffler 7 was way down on power in spite of some impressive claims being made by its manufacturer. Mufflers 4, 5 and 6 were all high-performance road mufflers. 6 was constructed of stainless steel with welded seams and no packing, so it would hold up well in race applications. Numbers 4 and 5, however, were 'S'-flow glass packs with rolled seams. These usually have a short life in race environments, but they can be cut open and duplicated in stainless with stainless packing, with good results.

Some tuners are surprised when they find that certain mufflers actually seem to give a power increase. However, be assured that mufflers do not increase power! When such a situation arises it is an indication that at certain places in the power band the engine is being over-scavenged, but the addition of a silencer is changing the engine's flow characteristics so less fuel-air mixture is being lost into the exhaust during the cam overlap period, hence power now goes up. When I see something like this, at the first opportunity I aim to test with a shorter-duration exhaust lobe or slightly less overlap. Alternatively, if it is a push rod engine I will change the exhaust rockers from, say, 1.6:1 to 1.5:1 and check the result.

Another benefit of reducing over-scavenging is that it can reduce exhaust noise levels, which if you are just over the limit may help you get through noise tests. Also, in endurance-type competition a reduction in over-scavenging will help lower fuel consumption, which could mean fewer fuel stops or a lighter fuel load.

CAT PROBLEMS AND MODIFICATION

When it comes to CAT converters the advice is to have a look at what is available, check the rules to see where it must be located and, if there is no specific placement rule, move it around while testing on the dyno. Note that in high performance and competition environments CAT flow performance must regularly be checked as many do not hold up well to extended wide-open throttle running. A collapsed CAT can easily rob a 300hp engine in excess of 20hp.

If the core of the CAT looks like this on the inlet side it is probably okay. A more reliable method of checking the CAT for blockage is to measure the backpressure before the CAT when it is brand new, and then at regular intervals afterwards. Do the check preferably on the dyno, but out on the road at full throttle in 3rd gear holding the car on the brakes at 5,000rpm is also a valid test.

Figure 9.8 CAT modifications improve flow

However, it is not just the honeycomb innards of the CAT that cause an exhaust flow problem. Even when the honeycomb is in good condition a poorly designed CAT will impede gas flow due to turbulence caused by the exhaust diverging at anything from 35° to 60° into the honeycomb, then converging again into the tailpipe at a similarly steep angle. To overcome this problem the CAT must have a gentle entry and exit taper. On the inlet side a taper of around 10–12° allows the exhaust gas to expand gradually to the full size of the honeycomb with minimal turbulence. Similarly, on the exit a taper of 12–15° forces the gases to converge into the tailpipe without unduly disturbing the exhaust flow. When modified in this manner ordinary street CATs will double their flow rate (Figure 9.8).

HEADER FINISH AND FABRICATION

When it comes to header design and fabrication I regularly see many errors, and very basic errors at that. Enthusiasts waste hours polishing the inlet tract to a mirror finish then throw on a header with welding 'dags' hanging into each tube where they are welded to the flange. A few minutes with a grinder or round file would rid the header of such impediments to good gas flow.

Header tubes overlapping the exhaust flange like this seriously limit gas flow

Figure 9.9 The header is an extension of the exhaust port

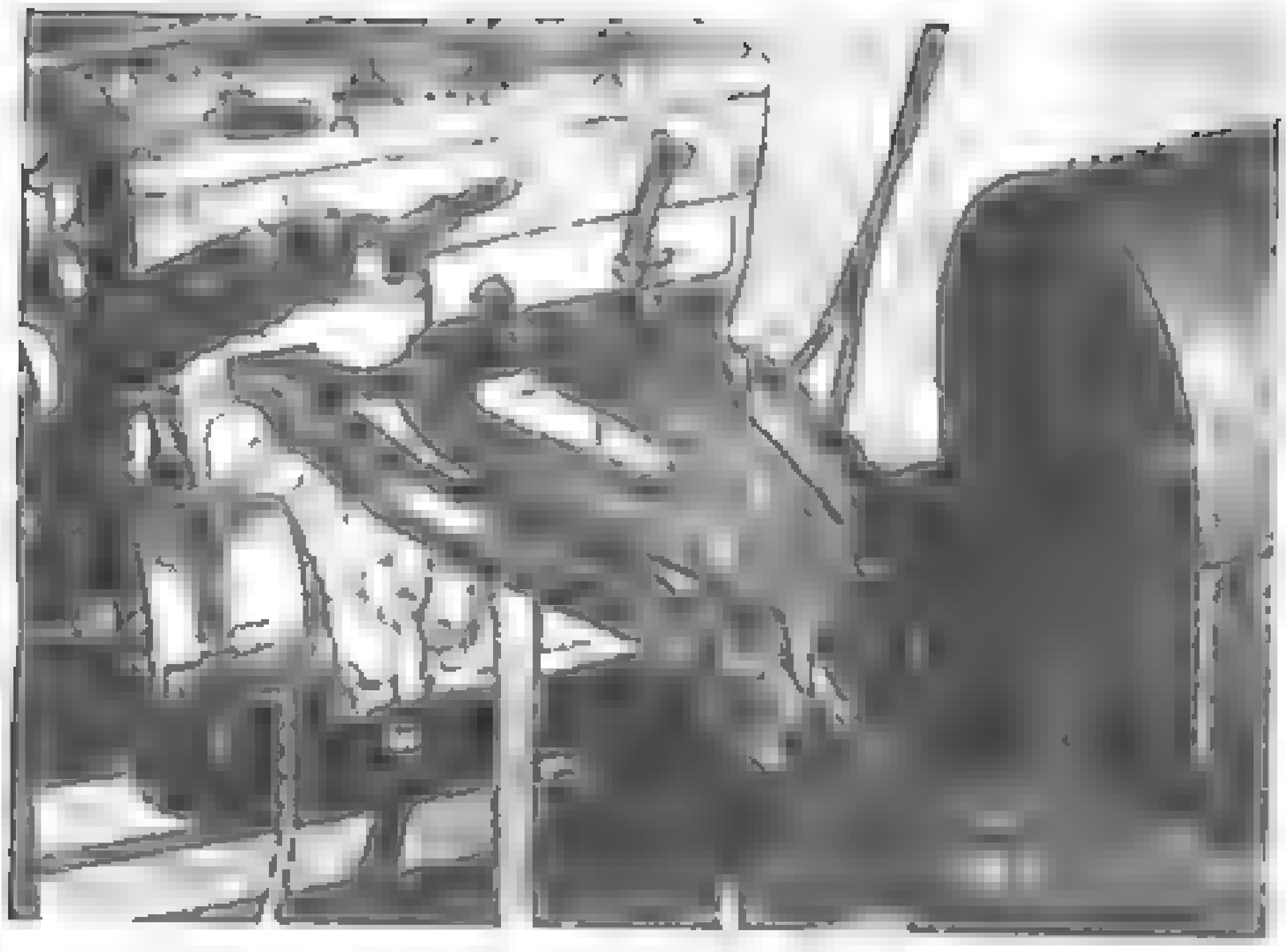
Another factor that many do not seem to understand is that the header is an extension of the exhaust port, so any abrupt change in direction as gas flows from the [illegible] as to avoid the time and expense of actually bending the tube (Figure 9.9).

To compensate for sloppy manufacturing and to ease fitting problems, grossly oversized flange holes are common, possibly allowing the header to overlap the exhaust port, thus restricting gas flow. This can be rectified by pushing sleeves into the oversize holes, then checking for overlap using the manifold gasket as a template

At times I do use slotted flange holes to check the effect of moving the header up or down relative to the exhaust port, usually a maximum of 0.125in. These slots are sized so that when the header is lifted to its limit, the floor of the exhaust port and header tube match. (This may improve peak hp a little.) Conversely, when the header is lowered the roof of the exhaust port lines up with the top of the header tube. (This can reduce exhaust reversion and improve mid-range hp.) I generally use this technique when doing header design work on the dyno, but on engines that show a liking for this sort of tuning it is a trick that can be used to 'tune' the engine to a particular circuit or surface (Figure 9.10).

Another problem area that has arisen since mandrel pipe-benders have become more commonly available to general exhaust workshops is a real lack of design planning in the placement and severity of bends in the header pipes, and indeed in the entire exhaust system. Years ago when most pipe-benders noticeably reduced the tube diameter at bends, a real effort was made to limit the number of bends and their severity. Just a quick glance at the exhaust told you that any sort of bend was bad news for gas flow. However, when mandrel benders came along the resulting bends looked so much better that many forgot that bends do impede gas flow. Also be aware that few so-called mandrel benders produce a perfect constant-diameter bend, many in fact reduce the pipe size by about 1mm.

Figure 9.10 Slotted dyno headers.



With wilder cams, testing of headers on the dyno is critical. Subtle changes can gain or lose a lot of hp anywhere in the rev range

THE CAM'S INFLUENCE

In some applications small tube headers are calle[illegible] but there is confusion over what to do when the exhaust port is larger than the header id. Obviously the lip formed when a 1.6in diameter port meets with a 1.375in id header tube will cause massive flow turbulence, but this is exactly what some small tube headers are like. Some tr[illegible] grinding out the flange to get rid of such a lip, but this does not really help very much. [illegible] a situation like this I prefer to flare the tubes gently over a length of at least 1in, and preferably 1½in. However, if the pipes have to bend very close to the exhaust port this may not be practical so it may be necessary to come off the ports in l[illegible] diameter tube for, say, 5–10in, then step down to the smaller tube size [illegible] flare the small tube up to the size of the larger over a length of 1–1½[illegible]

[illegible]n conclusion, let me make one final point that may save you [illegible] money and frustration. While we have had an in-depth look at head[illegible] [illegible]et too carried away applying this science to a mild street engine[illegible] header will work well with a road cam as pulse tuning is limited d[illegible] overlap period of this type of cam. A 3% power difference between an 'in tune' [illegible] 'out of tune' header is typical. However, when the valve overlap increases past a[illegible] [illegible] measured at 0.050in lobe lift, the header-tuned length becomes critical, to th[illegible] point that a slightly out of tune header can easily knock the power by 10[illegible]

I came across a dramatic example of this when setting up a 1600 VW Golf [illegible] engine that was supposed to have been an ex works unit. It made good power at [illegible] [illegible] end, peaking at close on 160hp at 7,250rpm, but there was a massive hole in [illegible] range; at 5,000rpm it showed only a little over 90hp when it should have been

Table 9.8 1.6 litre VW Golf header comparison

	Test 1		Test 2	
rpm	hp	Torque	hp	Torque
4,000	77.8	102.2	89.6	117.7
4,500	86.4	100.8	102.0	119.1
5,000	90.4	95.0	114.9	120.7
5,500	118.0	112.7	136.6	130.4
6,000	137.2	120.1	147.0	128.7
6,500	153.5	124.0	151.7	122.6
7,000	156.1	117.1	153.3	115.0
7,250	159.8	115.8	151.6	109.8
7,500	149.4	104.6	142.9	100.1
8,000	123.4	81.0	126.0	82.7

Test 1 – 4–1 Group A headers

Test 2 – 4–2–1 headers

least 105–110hp (Table 9.8). The owner was not overly concerned as he was regularly setting the pace in his class out in the forests. However, I was not convinced that the 4–1 Group A header was working, so we bolted on a 4–2–1 pipe, which I knew worked with the cam I regularly fitted in Golf rally engines. Power immediately rose to just under 115hp at 5,000rpm, while at the top end the engine was now peaking at 153hp at 7,000rpm. If the owner had had the money to outlay on further header development I am confident that we could have regained most of what had been lost at the top end while retaining the strong mid range.

Chapter 10

Camshaft and Valve Train

A camshaft and valve train assembly. Basically, the camshaft is designed to open valves before the piston starts a stroke and to close them after the completion of the stroke, in order to utilise the inertia or momentum of the fast-moving gases to fill and empty the cylinder efficiently.

The intake valve is opened before top dead centre (TDC) on the induction stroke, to get the valve moving off the seat before the piston starts down. It is then kept open well after bottom dead centre (BDC) to let the inertia of the high-velocity fuel/air mixture to literally ram in additional mixture while the piston is starting up on the compression stroke.

The exhaust valve begins to open long before the end of the combustion stroke. Most of the effective expansion power of combustion is over by mid-stroke and opening the exhaust valve early lets the cylinder pressure 'blow down' before the piston starts up on the exhaust stroke. By leaving the valve open after TDC the momentum of the exhaust gases is used to scavenge the cylinder efficiently. As the inlet valve is also open during this overlap period, the exhaust gas inertia will actually assist cylinder filling by creating a partial vacuum in the cylinder and inlet tract.

However, this theory is not going to work from idle to full engine rpm. At low engine speeds the fuel mixture coming into the cylinder has little velocity and consequently little momentum. In fact, the piston will start to push the mixture out of the cylinder back up the inlet port as it comes up on the compression stroke. A similar situation occurs with the late closing of the exhaust valve. When the outgoing exhaust gases have low inertia at low rpm, the piston travelling down on the intake stroke will cause the burned gases to turn around and be sucked back into the cylinder. The other possibility is the fuel/air mixture flowing straight past the exhaust valve during the overlap period. This can be particularly troublesome with hemi and pent roof combustion chambers. It is obvious that a compromise must be made to favour either low or high engine speeds. One cam cannot give you both with maximum efficiency.

CAM LOBE LIFT AND DURATION

Today, with the advances that have been made to improve the breathing ability of engines, it is not always necessary or desirable to employ a long-duration, high-lift cam to improve performance. In fact, such a cam can very easily spoil an otherwise well-thought-out engine modification. As we will discuss later, the valve timing and duration figures reveal very little about the power characteristics of a cam. The numbers game is played in the cam-grinding industry, too, so beware.

The science of camshaft design and operation is very complex, but our understanding of the basics will assist us to choose and correctly install a high [illegible] cam

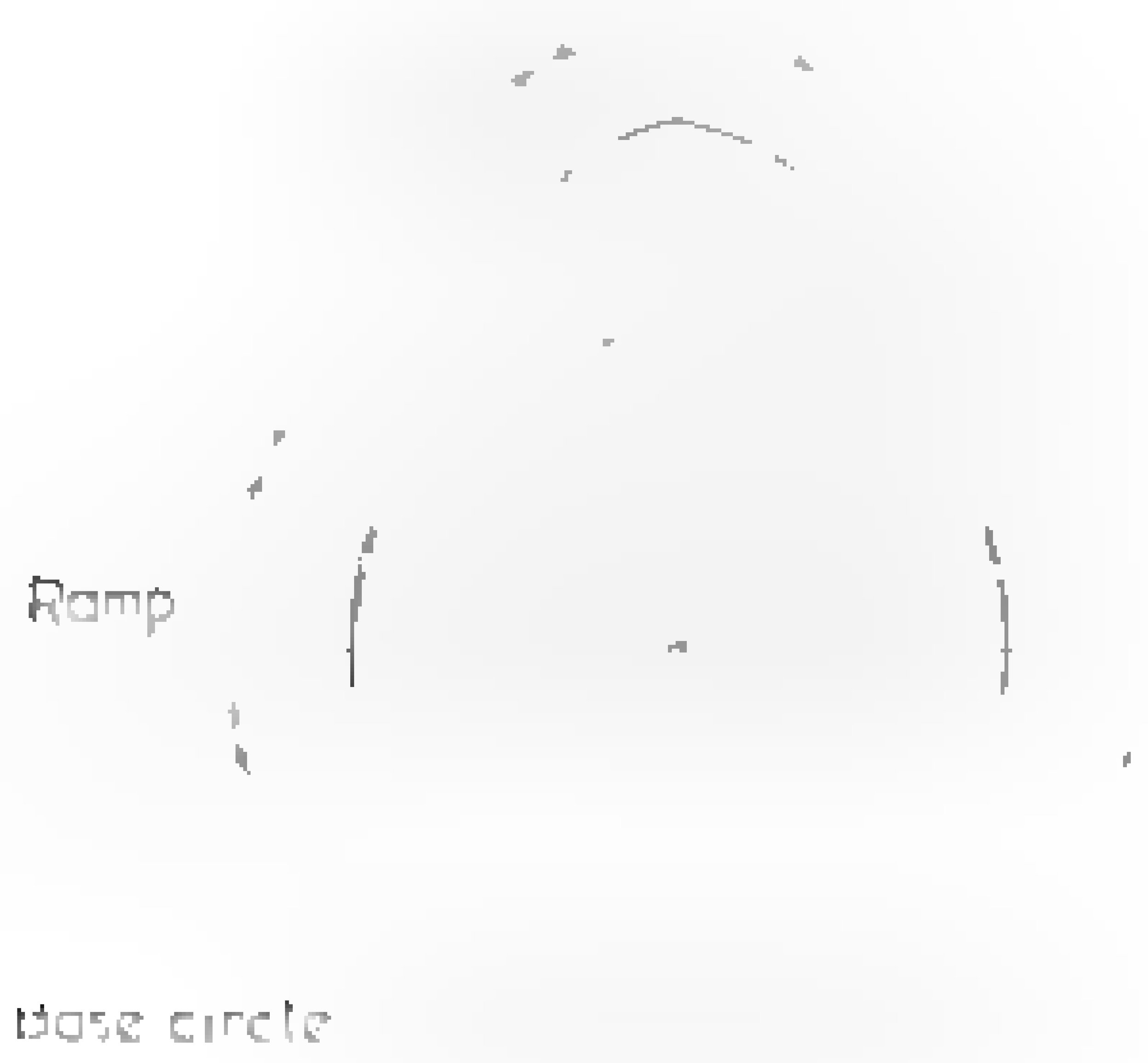
The base circle is the part of the cam that should, at all times, be at a constant radius from the centre of the cam core. The ramp (or clearance ramp) is the part that takes up the valve clearance and begins lifting the valve in a gentle manner, while the flank is the part that initiates the valve opening (Figure 10 1)

When designing a performance cam, the base circle must remain in the area of 140–160° (280–320 crankshaft degrees) This is necessary to allow the valves to dissipate heat and to give the whole valve train time to recover from the shock through which it has just gone. The ramp will, on a production cam, have 30–40° duration, while the flank will on average be 60–70°. To increase the duration of a cam we increase the flank angle to 70–80°, and to do this we must cut the ramp angle back to 20–30°

All production engine designers like to use fairly long ramps in order to lift and seat the valve gently. This has the effect of cutting down on mechanical noise and increasing camshaft life. However, when designing a performance cam we cannot reduce the base circle angle so we have to shorten the ramp

The average production engine timing is 10° 50°/50°–10°, ie the inlet valve opens 10° before TDC and closes 50° after BDC, and the exhaust valve opens 50°

Figure 10.1 Camshaft lobe.



before BDC and closes 10° after TDC. This type of cam will give good low-speed performance. A sports cam of around 25°–65°/70°–20° will improve performance with little loss in low-speed flexibility. The wildest cam I would recommend for a road machine is what I call a semi-race cam, which would have a maximum duration of 290°; the timing would be 40°–70°/75°–35°. After this we enter the field of full race competition cams, the shorter durations of which are more suitable for high-speed closed stage rally cars, road circuit cars and quarter-mile dirt speedway machines, while the longer-duration (320–330°) cams would be used in 1-mile speedway cars and drag machines. Table 10.1 indicates the power range of various camshafts.

Table 10.1 Camshaft duration and power range

Cam type		Rocker arm ratio			
		1.7:1	*1.5:1*	*1.25:1*	*1:1*
'Sports'	duration	198–214°	202–218°	208–224°	215–232°
	overlap	−20° to −6°	−18° to −2°	−12° to 8°	−5 to 16°
	valve lift	.450–.470"	.430–.450"	.360–.370"	.350–.420"
	power range	0.4–1.05	0.4–1.05	0.4–1.05	0.4–1.05
'Semi race'	duration	230–240°	235–245°	240–255°	245–260°
	overlap	10° to 25°	15° to 29°	20° to 40°	30° to 50°
	valve lift	.540–.560"	.510–.530"	.375–.400"	.410–.460"
	power range	0.6–1.1	0.6–1.1	0.6–1.1	0.6–1.1
'Full race'	duration	255–275°	260–280°	270–280°	270–285°
	overlap	43° to 65°	46° to 68°	54° to 73°	58° to 76°
	valve lift	.670–.850"	.600–.700"	.410–.430"	.430–.520"
	power range	0.8–1.1	0.8–1.1	0.8–1.1	0.8–1.1

Note: duration and overlap measured at .050in lobe lift
Valve lift with zero tappet clearance

To calculate approximate power range, refer back to Table 3.2 and formula in Chapter 3. Multiplying the rpm at which maximum hp is anticipated by the above figures will provide a guide as to the usable power range. For example, if the information in Chapter 3, pages 35 and 36 points to maximum power being produced at 7,000rpm and a semi-race cam is fitted, the power range should be about 0.6–1.1 x 7,000, or 4,200rpm to 7,700rpm.

The amount that the valves are lifted has a large bearing on the performance of an engine. Standard cams normally lift the valve about 23% of its diameter, while racing cams may increase this to 35% or more, even though flow in the inlet port may not increase, or even drop marginally when the valve is lifted more than about one third of its diameter.

Why then design a cam to lift the valve 35% of its diameter? It may sound silly lifting a valve so far, imposing higher loads on the valve train and making it necessary to use deeper valve cut-outs to clear the pistons, but this is how it works. Engine tuners have found that cams with quick opening and closing rates (high acceleration and high lift), but with relatively moderate duration and overlap, are a good way to get a broader torque curve. In other words, you pick up top end power without sacrificing so much mid range power. This occurs because the area under

the lift or displacement curve increases (see displacement curve in Figure 10.7) which improves the total quantity of flow into or out of the cylinder.

LOBE SEPARATION ANGLE

While many tuners spend a good deal of time agonising over cam duration and lift figures,

on different lobe centres affects the cam and how in turn this will influence the engine's power characteristics.

Simply stated, this angle fixes the actual position of the lobes on the camshaft

phasing to produce a valve timing figure such as 53°–77°/77°–53°, or 50°–80°/80°–50°, or 47°–83°/83°–47°. The inlet and exhaust lobes all have 310° duration, but grinding the lobes on a different centre has resulted in the inlet and exhaust valves opening and closing at different angles of crankshaft rotation.

Assuming that this 310° lobe profile is of symmetrical design, the lobe centre,

TDC on the induction stroke. For the exhaust we subtract the closing angle, in the first

Figure 10.2 Cam lobe phasing

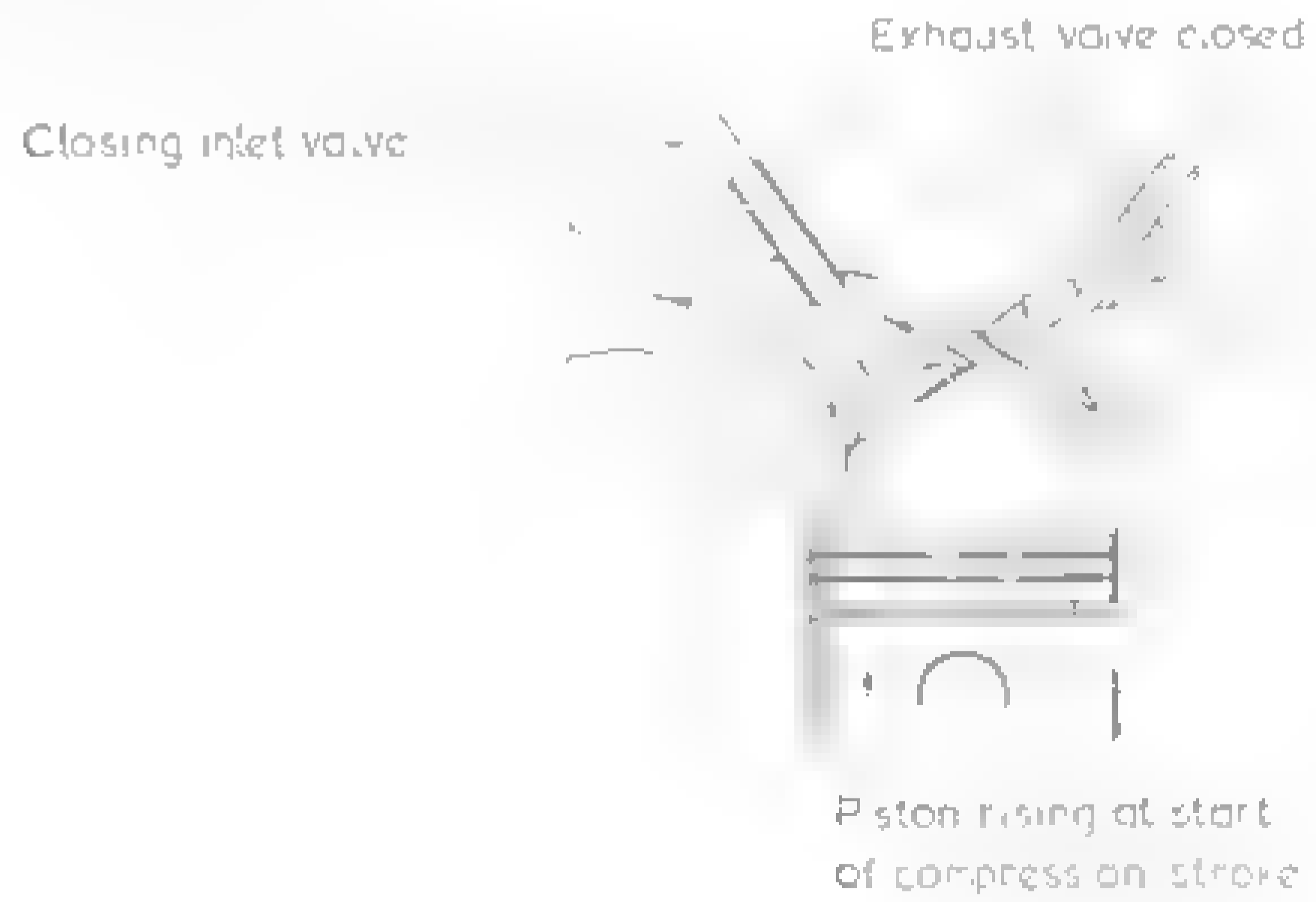
example 53°, from 155° and find the exhaust lobe centre to be at 102° before TDC. Hence this cam would be said to have 102° lobe separation. Following the same method you will find the other two examples have lobe centres of 105° and 108°.

On a small race engine of about 1.6 litres with good-size valves and ports and big carbs we might find that grinding the cam on 102° centres gives the best overall power spread. The inlet valve is closing sooner, reducing the amount of fuel/air mixture that will be pushed out of the cylinder by the rising piston (Figure 10.3). Thus cylinder filling will be better in the mid range of the power band, which increases hp. Also the increased valve overlap period of 106° (as opposed to 100° and 94° for the cams on 105° and 108° centres) should, in conjunction with the engine's good breathing potential, ensure that exhaust gas momentum has ample time to pull additional fuel/air mixture in, which will give good cylinder filling at high rpm.

However, a late-closing exhaust valve can cause problems at lower rpm. Because the outgoing exhaust gases have less inertia, the downward movement of the piston will readily draw spent gases back into the cylinder. These hot gases occupy a lot of cylinder volume, which limits how much fresh fuel/air mixture can be drawn in; they will also slow combustion of that mixture, reducing hp output (Figure 10.4). If this was the case, a lobe separation of 105° may work better.

When an exhaust port that is flowing well is connected to a properly designed exhaust header, another problem can be encountered. At higher rpm the exhaust may over scavenge the cylinder, drawing out not only all of the exhaust gas, but some fresh fuel/air mixture as well (Figure 10.5). The larger the cam overlap angle and the more efficient the exhaust port and header, the greater is the potential for this problem which robs the engine of high rpm power and fuel economy. The solution could be wider lobe separation angle or an exhaust lobe with shorter duration and/or less lift. Do not try to fix a problem like this by fitting a head with less efficient exhaust ports or a less efficient header. With a push rod engine you could swap the exhaust rockers to a lower ratio (say from 1.6:1 to 1.5:1).

Figure 10.3 Inlet reversion



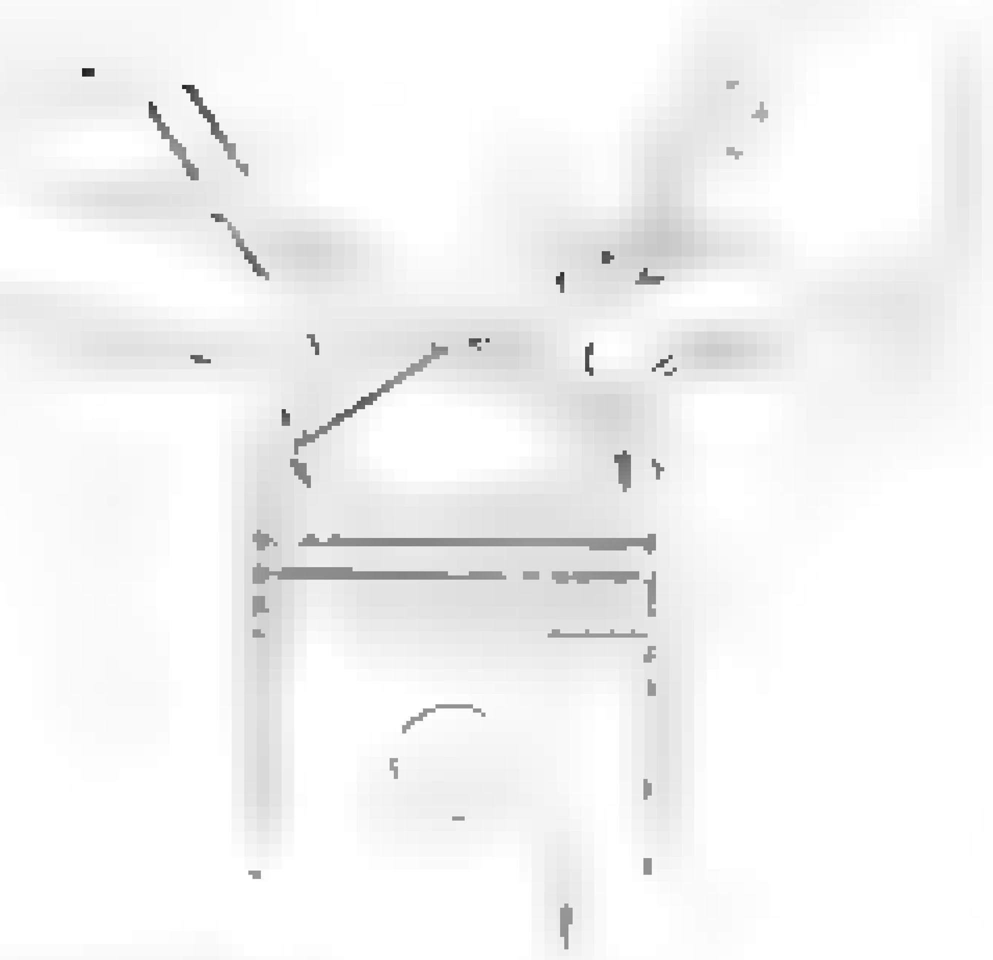
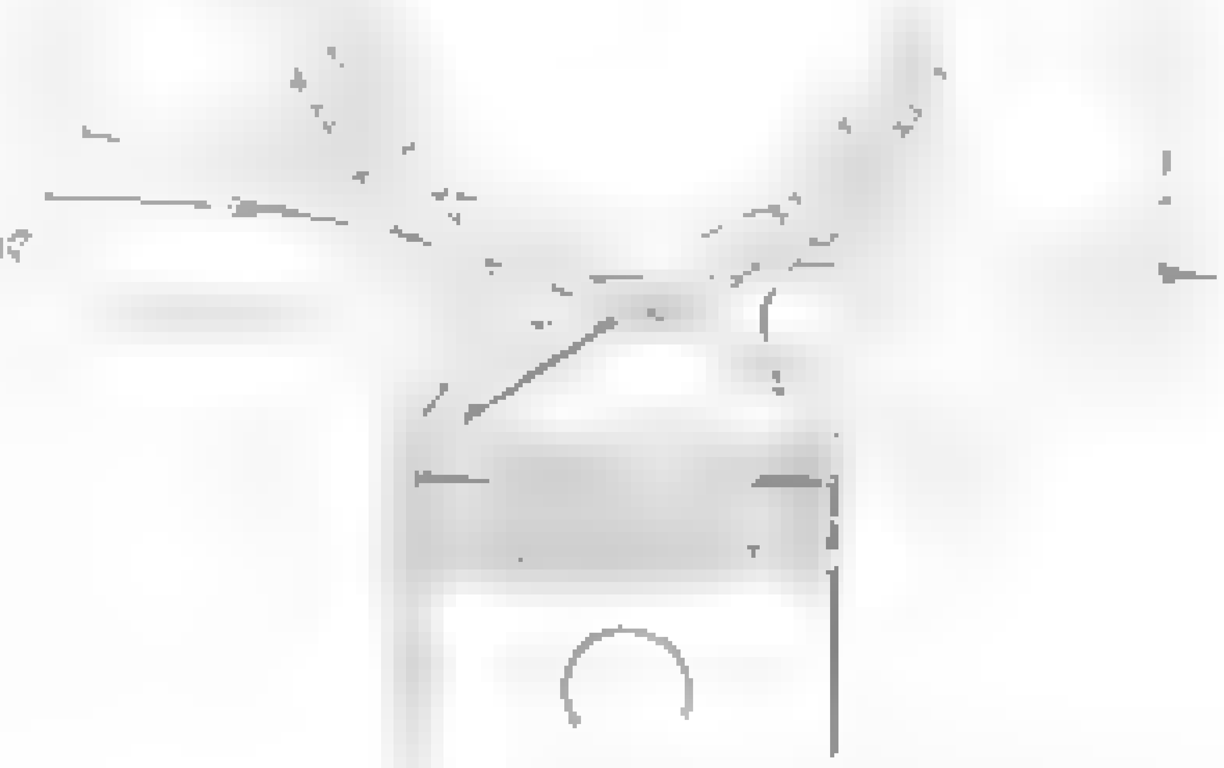


Figure 10.4 Exhaust reversion

In a race class that limits the engine to just one small carburettor (or inlet restrictors) and insists on stock inlet and exhaust ports, a cam with wider lobe centres may work best. The small carb and inlet port will ensure high gas velocity in the inlet tract, which will improve mid-range hp because of improved cylinder filling. Increasing flow velocity down the inlet tract also resists reversion flow back past the inlet valve as the piston rises at the end of the inlet stroke. Then at higher rpm the late inlet closing allows more time for the fuel/air charge to complete the cylinder filling. On the exhaust side, opening the exhaust valve early gives the cylinder more time to expel burnt gas through the tight exhaust port. Consequently less engine power is used to push the gas out past the open valve as the piston rises on the exhaust stroke.

Figure 10.5 Over-scavenging



Summing up, a narrow lobe separation angle tends to improve mid-range power, but go too far and bottom-end power could be lost due to exhaust reversion, and top-end could be lost due to over-scavenging. Conversely, a wider separation angle favours top-end power at the expense of a drop in the mid-range due to inlet reversion.

When we look at it from the aspect of port flow characteristics, we see that a comparatively restrictive inlet and exhaust calls for a wider phase angle. An engine with excellent inlet and exhaust flow, and a good balance between the two, will run a medium lobe separation angle (about 104–106° for race engines, 108–110° for road engines). However, an engine with good inlet flow, but by comparison only average exhaust flow, will want a narrower phase angle.

Obviously just changing the lobe separation angle may not provide the inlet and exhaust flow characteristics desired. What may be called for is a cam with more duration and lift on the inlet lobes, or we may need more on the exhaust lobes to get the best hp and a wide power spread. With other engines such as the Mini we may grind race cams with different lobe centres and even different lobes on the centre cylinders. On others such as a Chev V8 race engine with just one carburettor we do the opposite, giving the four outer (corner) cylinders more lobe duration and lift, or alternatively higher-lift rockers.

CAM ADVANCE AND RETARD

Another useful strategy to 'tune' the cam to give the desired power characteristics is camshaft advance, or in a few instances camshaft retardation. When we advance a cam with 108° lobe centres and 47°–83°/83°–47° valve timing by 6° we in effect advance the inlet lobe centreline to 102° after TDC, and likewise the exhaust lobe centre moves to 114° before TDC. The valve timing changes to 53°–77°/89°–41°, the inlet and exhaust valve both open 6° earlier and they also close 6° earlier. Mid-range power rises because reversion in both the inlet and exhaust tracts is reduced. Usually maximum hp falls a little. However, if with split timing (ie both lobes on 108° centrelines) the engine was over scavenging and drawing through the mixture, advancing the cam will reduce this problem, so maximum hp may improve a little (see Figure 10.12).

Basically, advancing a cam makes it work like one with narrower lobe separation, but without the disadvantages of the increased valve overlap and a late exhaust valve closing point. Thus a cam ground on 108° lobe centres and installed with 6° advance will perform in the mid-range like a cam with 104 or 105° lobe separation, and at high rpm will work similarly to a cam with 106 or 107° lobe centres.

Normally I do not install cams in the retarded position. Theory says that it causes the engine to lose mid-range power, but gain hp at maximum revs. I must have been doing it wrong all these years because any time I have tried retarding cams I lose power everywhere. In essence, retarding a cam, say, 4° will see the valves open 4° later, but more significantly they also close 4° later, which increases reversion in both the inlet and exhaust tracts and may also increase mixture draw-through on overlap.

A situation where we can take advantage of these normally unwanted characteristics is if the engine is detonating and the best fuel available is being used. For many tuners the first line of defence against detonation when the compression ratio has been made too high is to cut back on ignition advance. This is totally wrong – it kills performance. What we want is to reduce pressure in the cylinder. Cam retard

helps in two ways first it reduces cylinder filling at mid-range rpm with both inlet and exhaust reversion, and at higher rpm the late closing exhaust valve may permit some over-scavenging, which also reduces cylinder filling The second benefit is the result [illegible] This effectively slows combustion, therefore the combustion chamber, piston crown and valves all cool down, thus suppressing detonation

Certainly cam retard will cause some hp to be lost, but nothing like the amount caused by reduced spark lead. The amount of retard necessary will depend on how much too high the compression ratio is, and the cam lobe separation angle. Whereas 2–3° retard may get you out of trouble with a cam ground on 108–110° lobe centres, a narrower angle like 104–105° may require 6° or more retard

[illegible] around 9.1. These classes also often limit inlet breathing with a small carburettor or air restrictor. With a low-compression engine we have to come down about 6–10° on lobe duration to get the torque back, so if a typical short dirt-track motor on a 12:1 compression works best with 254° duration we would cut back to around 244–248° Then to achieve the best possible high-rpm cylinder filling, we would want a fairly wide lobe separation, usually about 110° with a two-barrel carb on a 350 V8, 108° with a [illegible] closing inlet valve killing mid-range power, so we dial in a lot of cam advance With 110° separation this will mean up around 8° advance; a long-duration cam may need another degree or two. A cam with 108° lobe phasing will probably want 7° advance.

If the rules limit compression to 9.1 but have unrestricted breathing, allowing a big carb or fuel injection, the situation is a little different. Again we reduce lobe duration, but the cam lobe centres will be narrower at around 106° as the engine has an unrestricted inlet system. Then to pump up the mid range the cam would be advanced probably 5°.

My experience over the years has been that regardless of whether the engine works best running a cam ground on 102° centres or 110° centres, all engines respond positively to some cam advance Race engines like to have the cam moved forward such that the inlet lobe centre is at 100°–102° after TDC. There are exceptions to this, but I do not remember very many Road engines run shorter lobe durations and have to get by with less fuel octane, so they typically give best overall performance with the inlet lobe centre around 104° 107° after TDC However, if you run into detonation at 107° and the maximum ignition advance has already been backed off 3 or 4° less than stock, you will have to go back to perhaps 109°. I do not like dropping down further than this as it just kills the mid range and leaves the engine really sluggish, fuel economy also suffers. If the engine is detonating with the inlet lobe at 109° and you have already pulled 3 or 4° off maximum ignition advance, you will have to lower the compression ratio or else increase the fuel octane

What if your race engine performs best with the inlet lobe centre at 104–106° rather than the 100°–102° that I have suggested to be the ideal range? The conclusion to be drawn is that the engine wants the inlet to close later, so probably the lobe separation angle is tighter than ideal, therefore the separation should be widened by a couple of degrees Alternatively the engine may want more lobe duration and lift on the current lobe separation angle. If it is a single-cam engine I would first try more duration and lift, but with twin cams it is easy and does not cost anything to swing the

separation angle apart. With the inlet in the 100–102° range, the engine should still make the same sort of mid range hp and the top end will now be stronger.

The other possibility for the engine preferring the inlet lobe at 104–106° is that this is reducing cylinder pressure and thus getting the engine away from trace detonation. Consequently, with a more controlled burn you are seeing a power increase. If there is a bore scope on hand, take a careful look at all of the pistons, particularly around the edges, for evidence of sandblasting. The other option is to lift the head and check for signs of detonation. If the engine is running into detonation, lowering the compression ratio and running the cam at the correct angle, along with correct ignition timing and fuel mixture, will improve performance and engine reliability.

UNDERSTANDING CAM DURATION FIGURES

Referring back to Table 10.1, you will note that all the duration figures are measured at 0.050in lobe lift. This serves to avoid confusion as people tend to measure duration in all sorts of obscure ways. Typically, car manufacturers measure duration from the point where the lifter commences to move. Some cam grinders measure the duration as starting when the valve begins to move taking into account valve train flex and valve clearance. Such a system appears reasonable until you begin to appreciate that clearances change with engine temperature and engine speed, as do valve train flex and camshaft deflection. Moreover, consider how stiffer valve springs alter these latter two factors. Other cam manufacturers rate their cams on a sliding scale. Soft road cams show a duration spec at 0.004–0.006in lobe lift, semi race at anything from 0.010–0.015in lobe lift, and race cams at 0.018–0.024in. All very confusing, I'm sure you will agree!

The solution has been to rate cams at 0.050in lobe lift. This is not perfect either, but it does give a better picture of what the duration number actually means and so gives some basis for comparing cams without actually testing every one. For example, looking at Table 10.2 you can see how dramatically advertised duration figures and [illegible] the same duration at 0.006in, but at other lift points cam No 1 has 11–19° less duration! Comparing No 4 and No 5 the same picture emerges, with the duration of No 5 being 12–15° shorter. However, when we look at the 0.050 figures we easily see that cams Nos 1 and 5 are both extremely mild road cams. Nos 2 and 3 are a bit warmer and should provide similar performance. Likewise with Nos 4 and 6, which are both very close to each other and hotter than the other four.

Table 10.2 Camshaft duration comparison

	No 1	No 2	No 3	No 4	No 5	No 6
Advertised duration (°)	236	252	260	260	264	266
Lobe lift (in)			*Actual duration (°)*			
0.006	254	254	255	260	266	260
0.050	195	206	204	212	200	210
0.100	163	175	174	181	168	[illegible]
0.200	96	115	114	121	106	120
Maximum lift (in)	260	285	284	292	272	[illegible]

Table 10.3 Race cam duration comparison

	No 1	No 2	No 3	No 4	No 5	No 6
Advertised duration (°)	290	294	291	300	299	298
Lobe lift (in)			*Actual duration (°)*			
0.050	260	261	263	270	271	268
0.100	229	226	225	231	230	237
0.200	183	179	174	184	182	191
0.300	137	132	125	136	132	145
Maximum lift (in)	.430	.418	.393	.425	.395	.435

The ideal situation would be for the cam manufactures to provide more detailed information, such as that presented in Table 10.3. If we only had the 0.050 duration figures we could mistakenly conclude that cam Nos 1, 2 and 3 are pretty much the same. No 3 has more duration than the others but it has less lift, so that should balance things out, or so it would appear. When we look at the 0.100 number we see a clearer [illegible] figures at 0.200 and 0.300in lift. The same could be said of Nos 4, 5 and 6. At the 0.050in lobe lift figure they are all around the 270° mark. No 5 has less lift but it has a few degrees more duration, but at 0.100in it shows itself to be a significantly milder cam.

Does this mean that if you were looking for a 260 or 270° cam, Nos 1 and 6 would get the nod? Well that depends – I would be worried about their more aggressive profiles killing the valve gear. That may not be a huge concern in a sprint-type engine used in hillclimbs, for example, but how about if we were running a distance endurance event? How long would the valve springs and rockers, not to mention the valves and valve seats, stand up under such a hammering?

VALVE BOUNCE AND TOSS

Added to these questions, all of which demand answers, it is sobering to remember that a more aggressive profile on paper may not in the event deliver in practice when installed in a race engine. Figure 10.6 illustrates what may actually be occurring at the [illegible] train flex and the lifter faithfully following the profile of the cam lobe, finally easing the valve down on to its seat. The other curve depicts actual valve motion during the first 100° [illegible] the lobe attempts to decelerate the valve as it approaches the point of maximum lift, the valve train and valve mass overwhelm the valve springs and the lifter is tossed off the lobe. For the next 160° the valve continues to act independently of the cam lobe. At this point valve spring pressure and valve and valve train mass combine forces to slam the lifter back on to the face of the cam. This combined mass then causes the entire valve train to bend again, allowing the valve to crash on to the valve seat around 25° ahead of schedule. However, rather than remaining seated the valve bounces off the seat and does not reseat for another 80°, and finally settles after a further 130°.

[illegible] sobering to reflect on the fact that such moves may reduce the amount of 'toss', but the [illegible]

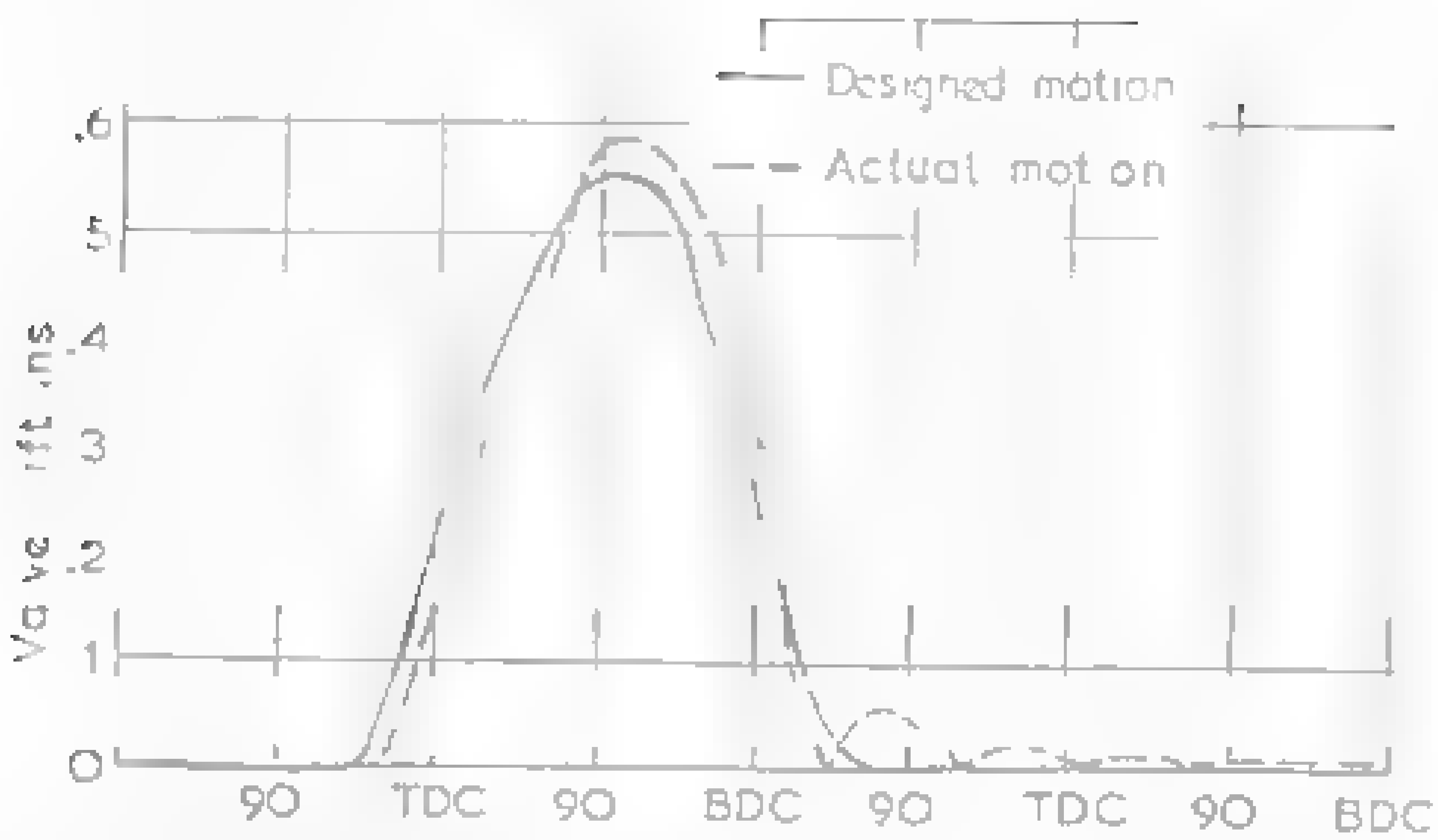


Figure 10.6 Valve motion curves

bounce on to the seat may in fact worsen. The real solution may be a more gentle cam profile or a lower maximum engine speed.

It should be obvious that from a mechanical point of view valve bounce is to be avoided, but additionally it is something we must eliminate with respect to engine hp. Clearly if it is the exhaust valve that is bouncing off its seat, the intake charge will be contaminated by exhaust gas drawn back into the cylinder past the bouncing valve. Then, as the inlet valve bounces off its seat, fuel/air mixture is expelled by the piston rising on the compression stroke. This lowers cylinder pressure and shoots a massive reversion blast up the inlet tract, which in turn will disrupt the following inlet cycle.

Thus in the real world cam profiles Nos 2 and 4 may assist the engine to generate the highest hp because they maintain better control of valve train motion. However, cam Nos 3 and 5 may indeed be the ideal choice. They may be down a touch in the power stakes, but they could be much more gentle on the valve gear. Also, on the track they may prove to be faster because they will allow the engine to safely rev faster by, say, 300rpm without valve bounce.

Valve toss isn't always bad from the standpoint of max hp in some race engines. The specific exception is with regard to silly regulations limiting maximum valve lift. The original intent of such rules was no doubt noble, penned to keeps costs down and even out the competition. Racers are always looking for the winning edge so those with more money approached engine tuners looking for a legal way around this rule. The solution was to design the lobe and valve train to deliberately toss the lifter off the nose of the cam, effectively gaining valve lift, and thus more hp and rpm. This isn't kind to the valve train, but when valve motion is controlled so that they don't bounce on the seats, it provides a winning advantage.

CAM LOBE DYNAMICS

This leads us to what we call cam dynamics. Figure 10.7 shows a displacement curve, a graph of camshaft rotation in degrees relative to the motion of the cam follower. Y

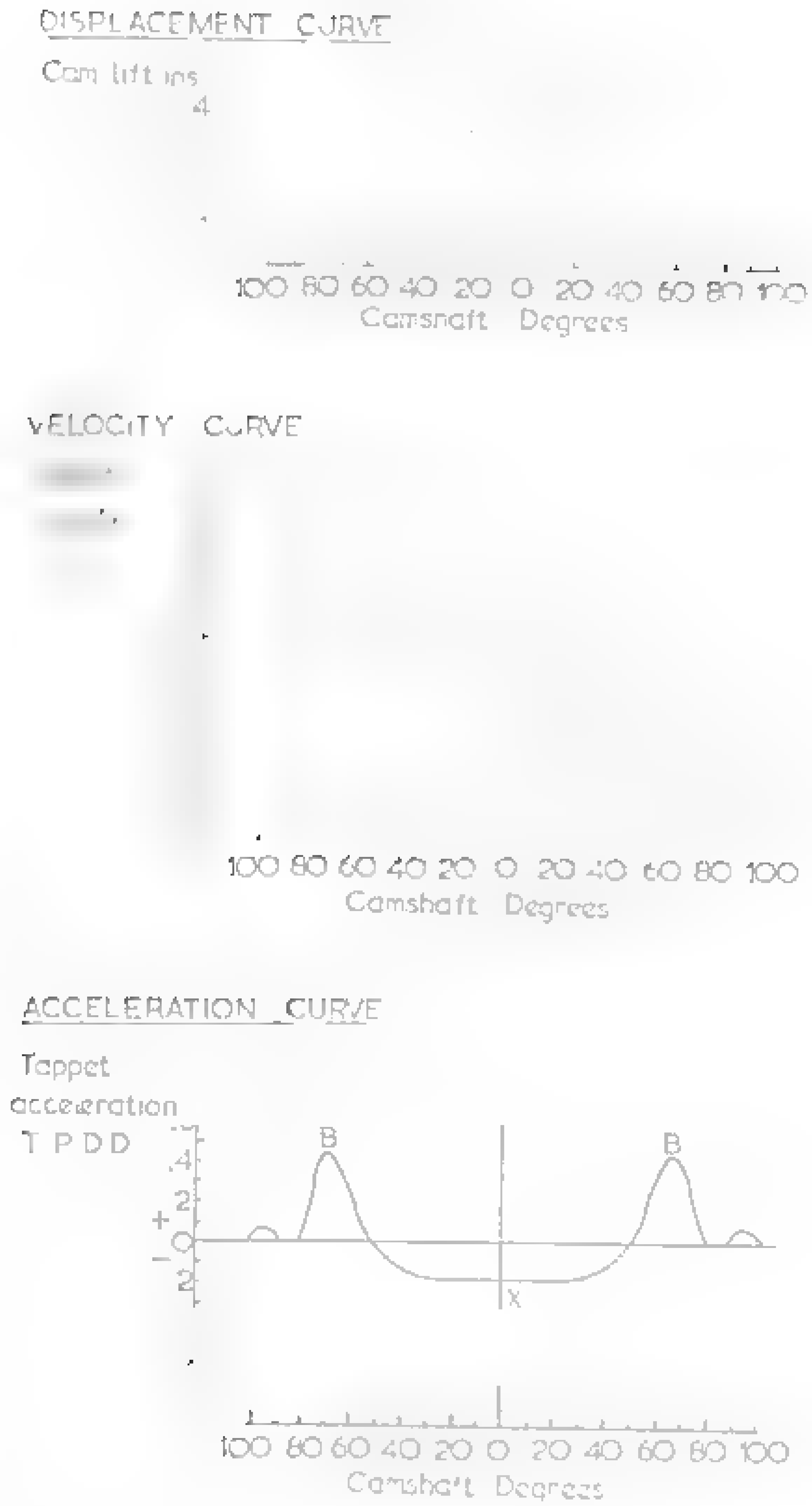


Figure 10.7 Cam lobe characteristic curves

can readily see that just knowing the duration and lift of a cam will not give a true indication of its performance potential. The dotted line shows a cam of the same timing and lift but with quicker opening and closing times.

The velocity curve is the rate of lift expressed in thousandths of an inch per degree of rotation. The most significant parts of this diagram are the maximum velocity points. The maximum velocity of a cam occurs when the cam is contacting the follower nearest its edge. With a given tappet diameter, a maximum practical cam velocity can be used without the cam striking the sharp edge of the cam follower. Table 10.4 indicates the maximum velocity for various cam follower diameters. Most high performance cams produce a velocity of 0.007in per degree (or 7 thousandths of an inch per degree), which is generally within limits, but some racing cams are running 0.009 and up to 0.012in per degree velocity, which is way above the safe limit for many engines. Once the safe limit is passed, high cam wear and follower breakage is imminent.

Table 10.4 Maximum safe cam and tappet velocity

[illegible] diameter (in)	[illegible] velocity (TPD)	Engine application
0.780	6.63	
0.800	6.8	Ford Anglia/Cortina
0.812	6.9	Austin 'A' Series
0.842	7.16	Oldsmobile 260/455, Chevy 6 and V8, Holden 6
0.850	7.23	Cortina/Escort (Kent motor)
0.874	7.43	English Ford 3000 V6, American Ford 6 and V8
0.904	7.68	American Motors 6 and V8, Chrysler 6 and V8
0.921	7.83	Oldsmobile 1967 model 400/425
0.941	8.0	VW Type 4
0.960	8.16	Replacement Chevy mushroom tappet
0.980	8.33	
1.000	8.5	Replacement Ford V8 mushroom tappet
1.060	9.01	
1.100	9.35	
1.125	9.56	
1.155	9.82	VW
1.186	10.08	Morris 1,500 and 1,750 ohc
1.200	10.2	
1.300	11.05	
1.375	11.69	Lotus/Ford twin cam, Jaguar, Alfa Romeo dohc
1.400	11.9	

The acceleration curve shows us the rate of change of velocity in inches per degree, per degree. As will be explained when we deal with cam 'fingerprinting', this curve is critical in showing factors that affect reliability, as well as information concerning the maximum rpm attainable by the valve train. Negative acceleration determines this maximum rpm. This is the part of the valve train motion that is working against the valve spring, as the lifter is slowing down to go over the nose of the cam. As the engine turns faster and faster, a point is reached where the force generated by the mass

of the valve train equals that generated by the valve spring in the valve open position Any further increase in rpm would cause valve float. A cam profile with the lowest negative acceleration (or deceleration) will allow the highest rpm potential or allow the use of softer valve springs if that rpm potential is not going to be used. Ideally, negative acceleration should be less than –0.0002, but up to –0.00028 is tolerable (–0.2 to –0.28 thousandths of an inch per degree, per degree)

The two higher-positive acceleration peaks are significant in assessing the durability of a cam, since a heavy valve train combined with high acceleration will [illegible] low, but stresses due to inertia are high The two smaller positive acceleration peaks [illegible] 0.0006–0.0007in per degree, per degree (0.6–0.7 thousandths of an inch per degree, per degree).

To help you understand what you are looking at, I have marked some points of significance on the three curves. Point 'X' is the nose of the cam, where full lift occurs Point 'A' is where maximum velocity is reached, and 'B' indicates the point of maximum acceleration. 'A' is also where cam lobe contact is at its farthest from the centre of the tappet We call this the point of maximum eccentricity

MEASURING CAM CHARACTERISTICS AND ACCURACY

Quality and grinding accuracy are both very important considerations in buying a high-performance camshaft, but unfortunately they are not always found. An inaccurately ground cam will lose you a lot of power, cause tuning problems, inducing early valve float and causing premature lobe and lifter wear. Fortunately, there is a way of not only checking out the cam, but also delving into its actual profile, to produce the curves shown in Figure 10.7. All that is needed is to set up the cam in a lathe or rest it on a set of V-blocks. A 0.5in dial indicator is then attached to one of the lobes and a degree wheel bolted to the cam. I always check my racing cams on the two lobes in the middle, and also No 1 intake and exhaust lobe for comparison. If the cam is bent, or if it has been forced against the grinding wheel, this will show up more on the centre lobes. When I find a cam to be outside the tolerances mentioned, I return it to the grinder for replacement

With the cam set up so that it can rotate between centres, and with the 360° degree wheel bolted up, set a fixed pointer next to the degree wheel and position the dial indicator to read off the desired lobe. It is best to make a fixture to hold the actual lifters you will use with that cam and set up the dial indicator to read off the lifter. For most purposes you can check the cam every 0.005–0.010in of lift

It is very important to set up the dial indicator accurately and solidly. With this done, and the cam rotated so that the lifter is on the base circle, zero the dial indicator Rotate the cam to lift and lower the cam follower at least six times to verify that the indicator returns to zero with the follower on the base circle. If it does not return to zero either the lifter is sticking or the indicator is not mounted solidly.

The first thing we want to check is the base circle run-out This should be around 0.001in, certainly more than 0.002in is unacceptable. Next we check the actual timing and lift The cam grinder should indicate how he arrives at his claimed timing figures Some measure the duration as starting and finishing when lift is 0.020in, and others

measure a few degrees past the end of the clearance ramp, where lifter acceleration is positive. As already mentioned, many are now specifying a timing figure for 0.050in lift, which is better as we are well off the ramp at this stage, so the figures are more accurate. I work to a tolerance of ±1° of the stated opening and closing points, and ±0.002in of the rated lift. Remember that the cam duration is only half of the advertised duration, which is measured off the crankshaft.

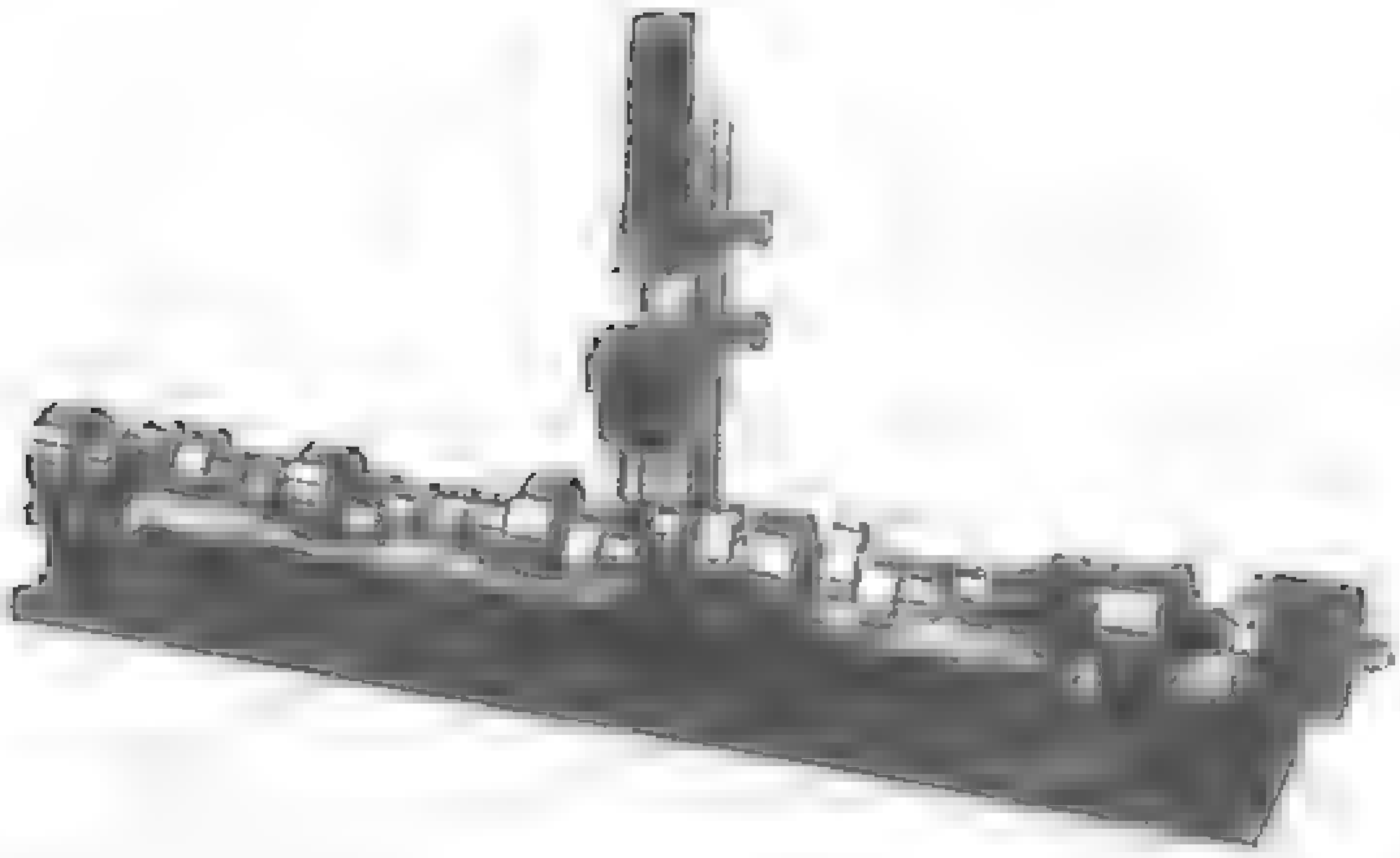
After these basic checks we get down to the nitty-gritty of profile curves. As shown in Table 10.5, we record the cam angle for every 0.010in lift. Once we have a full set of figures for that lobe we can set about computing the velocity and acceleration figures. You will note that these figures are placed between the basic lift and angle reading. The third column will be the change in degrees for each 0.010in lift (Δ). The fourth column is lift velocity expressed in thousandths of an inch per degree of rotation (TPD). To find the TPD figures simply divide each Δ° into 10. The fifth column is the change in TPD (ΔTPD). This tells us the amount that the velocity of lift is changing for each 0.010in of lift. The final column is for acceleration, which we call TPDD or thousandths of an inch per degree, per degree, and is worked out using the following formula:

$$TPDD = \frac{\Delta TPD}{\Delta^\circ}$$

The acceleration figures show us the accuracy of the profile. If the figures jump all over the place this may indicate grinding wheel chatter marks, or it may indicate polydyne correction in the cam design. The machine that grinds cam lobes traces the pattern from a larger master cam. The grinding wheel is controlled by a spring-loaded roller that rides on the master cam as it rotates. If the operator feeds the grinding wheel against the cam too rapidly, the roller can be forced off the master cam, resulting in a cam profile that does not conform to that of the master.

Table 10.5 Cam profile table (in part)

Cam lift (in)	Angle	Δ°	TPD	ΔTPD	TPDD
0.010	101.6°				
		7.6	1.32	0.90	0.118
0.020	109.2°				
		4.5	2.22	0.72	0.164
0.030	113.7°				
		3.4	2.94	0.51	0.150
0.040	117.1°				
		2.9	3.45	0.55	0.189
0.050	120.0°				
		2.5	4.00	0.76	0.304
0.060	122.5°				
		2.1	4.76	0.80	0.381
0.070	124.6				
		1.8	5.56	0.69	0.383
0.080	126.4°				



Cam lobe characteristics and camshaft accuracy can be assessed using fairly basic equipment. This takes more time than with more sophisticated gear that inputs directly to a computer, but is no less accurate

Polydyne correction is quite different, and is a designed correction factor that has been planned to correct some valve train problem caused by the fact that most valve trains are not rigid enough (due to push rod flex, camshaft flex, etc). What this correction really does is to accelerate the lifter, then drop back, then accelerate again. This is done to give the valve train a chance to 'catch up' so that it is not overstressed. If you find your velocity and acceleration figures are regular and consistent with, say, four or six of these bumps in one full rotation, you can assume that the cam profile has polydyne corrections.

To help in determining this I always draw up an acceleration curve. This will help determine if the 'bumps' are to a set pattern. A golden rule of cam design is that the area inside the acceleration curve on the positive side must always equal the area inside the curve on the negative side. It is quite logical: if you are going to accelerate something a certain amount, the only way you are going to get it back to a standstill is to decelerate it by the same amount. If the cam does not conform to this rule, valve train problems will result.

The next problem takes some time to check as we must check each lobe to determine correct indexing. If the cam lobes are not phased (or indexed) at exactly the right angle to each other from cylinder to cylinder, the valve timing of the whole engine will be amiss. I have seen cams with lobes out of phase by up to 15°. When checking the phasing I also record the maximum lift for each lobe. As mentioned earlier, I would expect the lift to be within ±0.002in of the cam grinder's quoted figure, and all the lobe centres (full lift angle) should be in phase by ±1°.

This means that a four-cylinder engine will have all four inlet lobe centres 90

(±1°) apart. Therefore if the firing order is 1–3–4–2 and No 1 lobe centre is at, say, 70° on the 360° degree wheel, No 3 should be at 70° + 90° = 160°, No 4 at 160° + 90° = 250°, and No 2 at 250° + 90° = 340°.

The four exhaust lobe centres should also be 90° apart, but to determine at what angle they should be in relation to the No 1 inlet lobe, we have to go back to the cam timing figures and do some calculations. If the cam had symmetrical timing of 30°–70°/70°–30° this would mean that the valve open period would be 30° + 180° + 70° = 280°, measured at the crankshaft, or 140° at the cam. Therefore the inlet lobe centre should be at

$$\frac{280}{2} - 30° = 110° \text{ ATDC (at the crankshaft)}$$

and the exhaust lobe centre should also be at

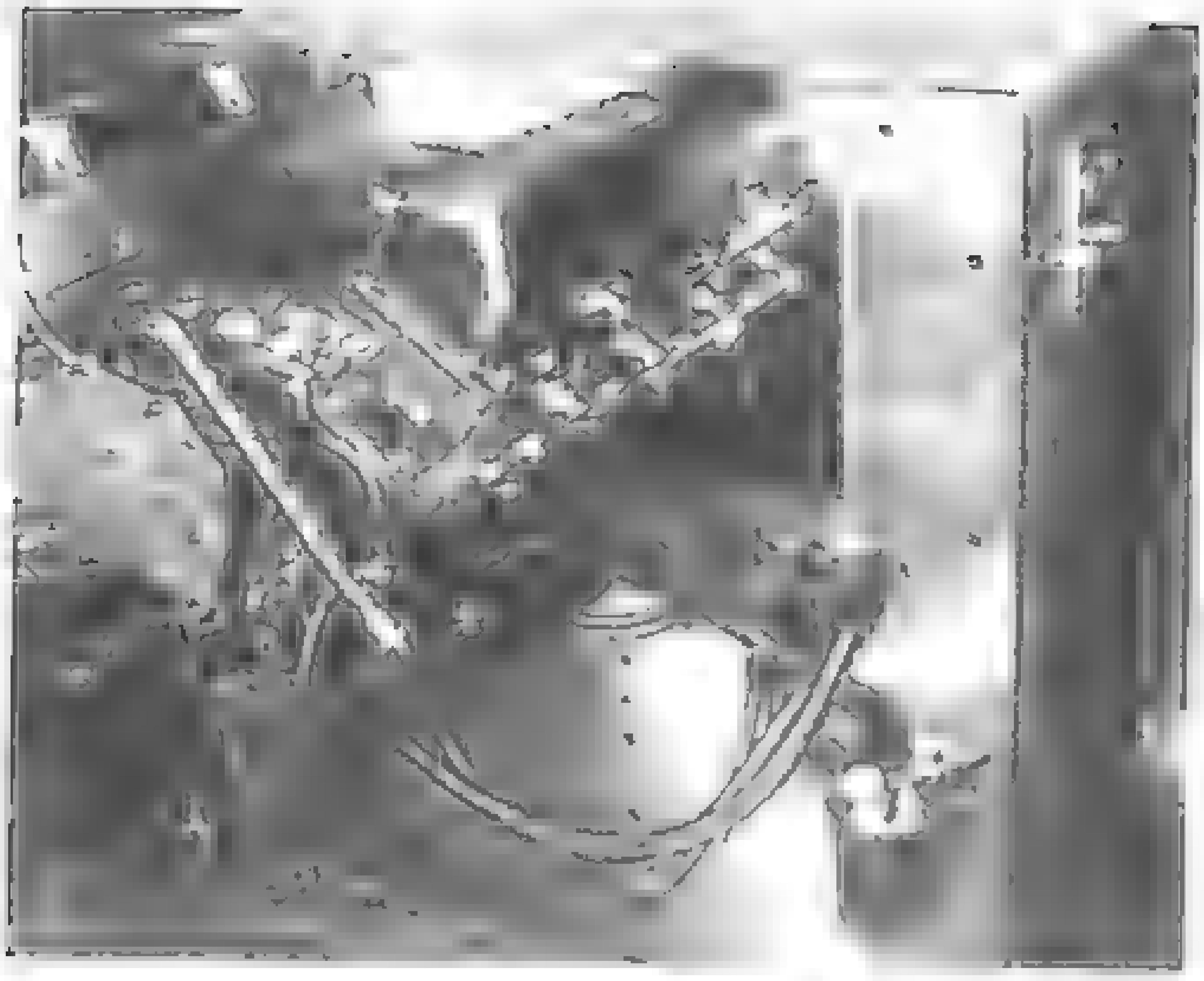
$$\frac{280}{2} - 30° = 110° \text{ BTDC (at the crankshaft)}$$

At the camshaft the No 1 exhaust lobe centre should be

$$\frac{110° + 110°}{2} = 110° \text{ ahead of the No 1 inlet lobe}$$

This would mean, using the above example, that the lobe centre for No 1 exhaust would be at 70° – 110° = 320° (±1°) on the 360° wheel. No 3 would be at 320° + 90° = 50°; No 4 at 50° + 90° = 140°, and No 2 at 140° + 90° = 230°. It all looks very complicated on paper, but once you sit down and work it out with a cam and a degree wheel in front of you, it is no problem at all.

Note that there are exceptions where the lobe phasing and/or lobe profile may be deliberately altered on specific cylinders for certain engine types; as previously mentioned the 'A' and 'B' series Austin engines with 5 port head and push rod American V8s, running in demanding classes of competition. Obviously we want every cylinder to produce, ideally, identical hp. The obstacle to achieving this may be race rules limiting the type of cylinder head, inlet manifold and carburation permitted. Alternatively the engine may have a basic design limitation and competition is just so intense that we have to venture into ridiculously high engine speeds, aggressive lobe profiles and valve spring pressures to stand any sort of chance of being a winner that really weird things begin to occur with regard to actual valve motion. In push rod V8s the problem is a single camshaft confronted by massive valve spring loads from sixteen big heavy valves, being banged open perhaps 0.750in at engine speeds approaching 10,000rpm. In addition to the actual valve timing at the valve seat being muddled by flexing of the push rods and rockers, the camshaft will begin to wind up as engine rpm increases causing the lobes furthest from the drive sprocket to lag behind. The extent of valve train flex and lobe lag is checked in a Spintron at various rpm, and then, depending on where in the engine's rev range we need to pick up more power, the lobe phasing will be progressively advanced along the camshaft. Obviously this takes many experimental cams and a lot of dyno time to get right.



... checking of valve ... stability etc. with laser measuring devices. High speed video and ... hy may also be employed

THE CAM FOLLOWER

The next vital link in the valve train is the cam follower (also called tappet, lifter or bucket). This has the task of changing the rotating motion of the camshaft to an up-and-down motion. The contact point between the tappet and the cam lobe is the most heavily loaded spot in any engine; it can be as high as 300,000lb/in², which is one reason why ... followers require very special attention in a high performance or racing engine.

Contrary to popular opinion, cam followers are not flat; in fact, they are ground with a spherical radius of 37-75in, which means that they are around 0.003in high i... ...re. This spherical radius, along with a cam lobe angle of 3-16', and the tapp... ...re of the lobe, causes the tappet to rotate (Figure 10.8). The effect ... and tappet wear; in fact, if the tappets do not rotate, this part of your trick engine is quite likely to self destruct in around 15 minutes. For this reason I always have the cam followers ground to the required spherical radius whether they are new or used. I have found many new followers to be flat, which, while not bein... ...s old followers worn concave, means th... ...ot up to the task of pro... reliable service with a performance cam.

...re we take a look at roller tappets it might be ...sed to examine ... types of flat tappets. You will note that I make no mention of the hydraul...

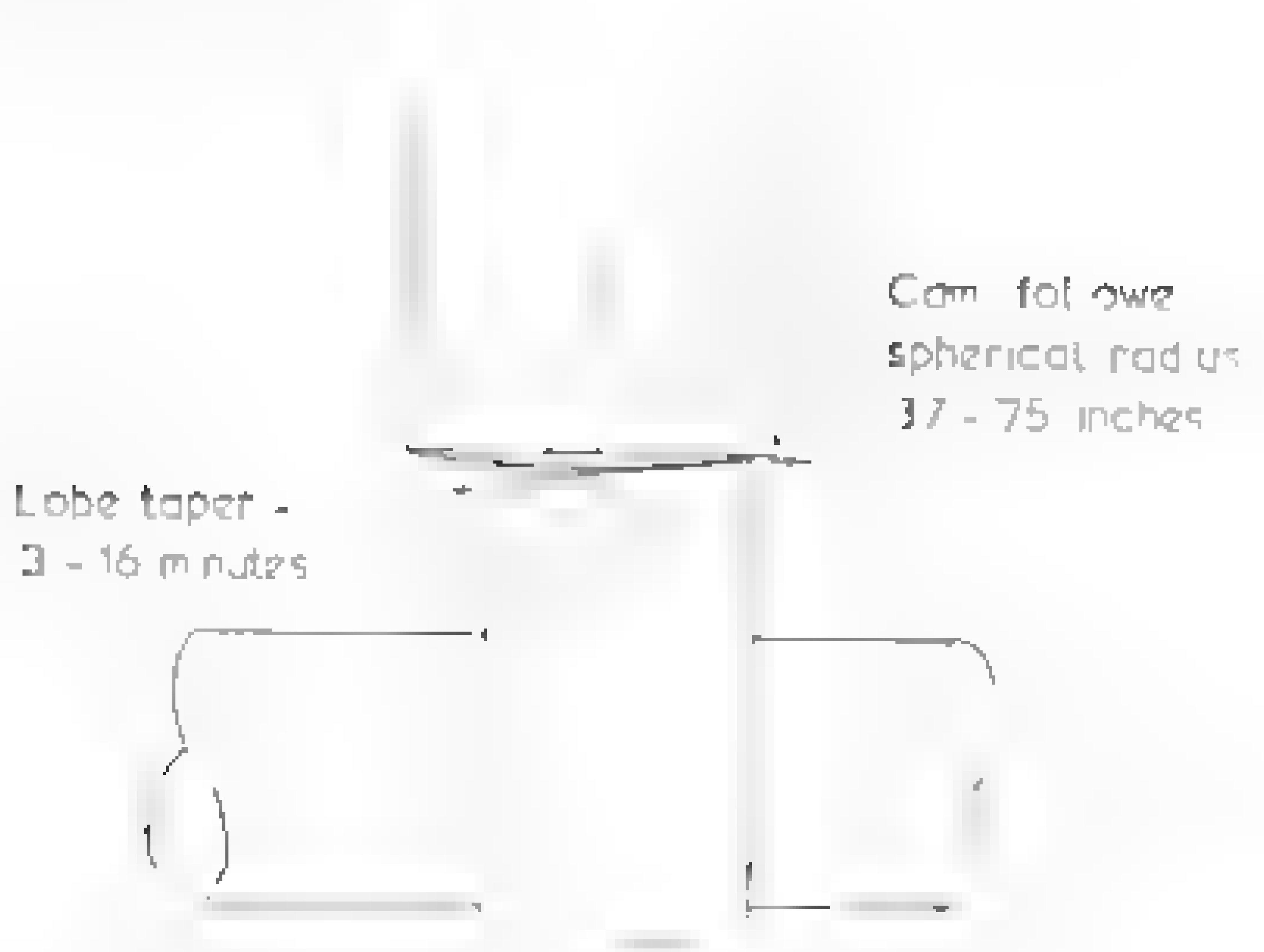


Figure 10.8 Cam lobe and tappet design

because in function it is similar to a conventional flat tappet. The conventional bucket type flat tappet used in most double-overhead-cam and a few single-overhe engines has many advantages: light in weight, large in diame and with a small rocking moment. However, manufacturers and service mechanics do not like this simple arrangement because of the tedium of adjusting the valve clearance. This

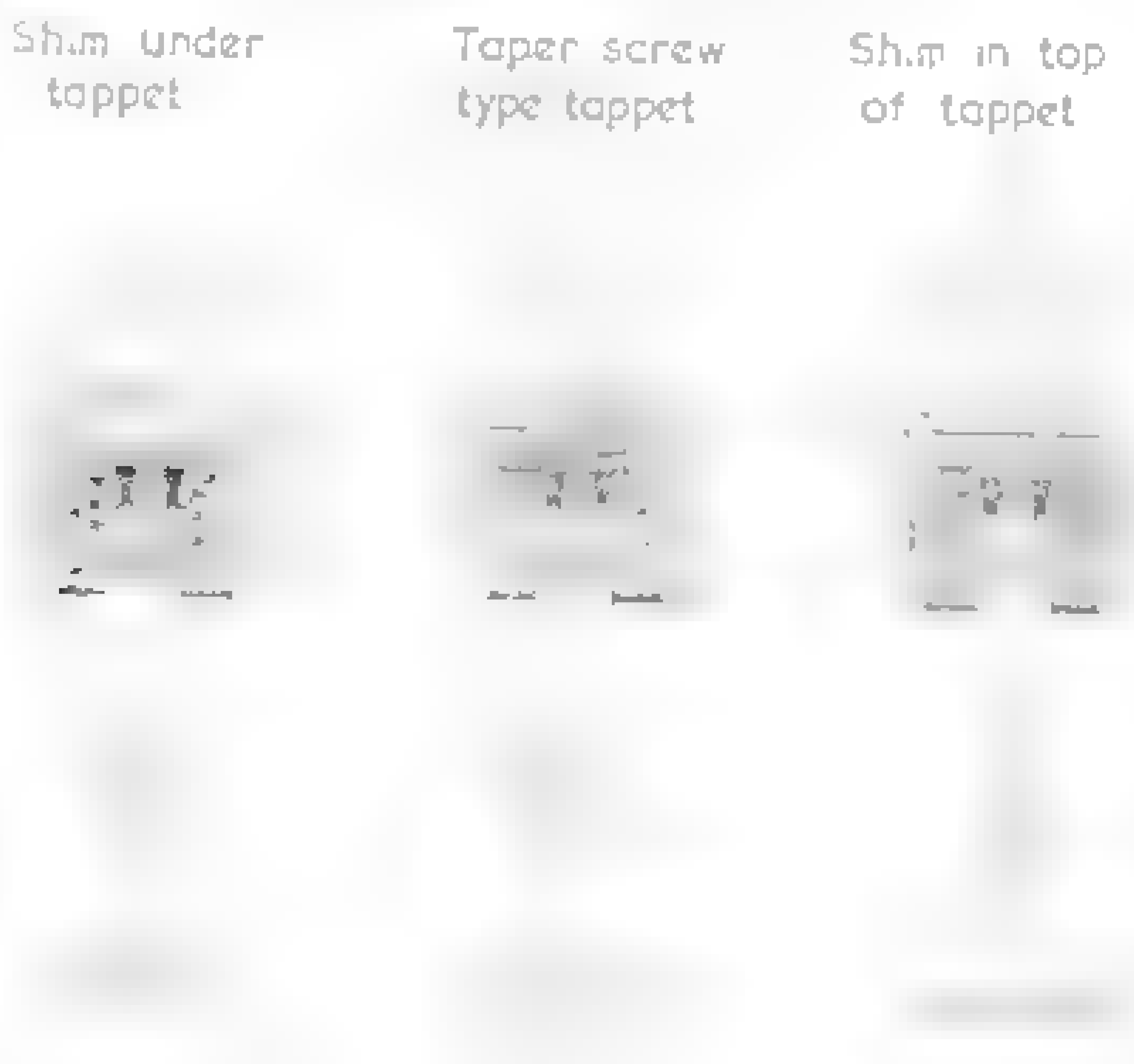
adjustment is made by inserting shims of various thicknesses between the valve stem and bucket (Figure 10.9). This means that the cams have to be removed each time valve clearance adjustment is needed, then, on being refitted, they have to be re-timed.

Manufacturers have devised a couple of ways to get around this problem, but in [illegible] Vega both used an adjusting screw with a tapered flat on one side, which does the same job as the adjusting shim. Using an Allen key, the clearance is adjusted by turning the screw in or out. Adjustments must be made in full turns as the screw has a flat only on one side.

The system most often used has a shim recessed in the top of the tappet. To adjust the clearance, a special tool is used to compress the valve spring. The shim can then be flicked out of the recess and another shim of the required thickness fitted.

While both designs have simplified valve adjustment, the weight of the bucket tappet has in some cases been doubled, as compared with the conventional bucket tappet and shim. This is not such a problem on more mundane motors, but high-revving motorcycles and race engines are a different matter. The extra weight of the valve train imposes a higher load on the valve springs, and this lowers the valve float rpm drastically. To combat this, valve spring pressures must be increased; however, an increase in spring pressures soaks up power and increases valve train flex and wear. Another problem is that of missed gear changes or deceleration over-revving causing valve float. This then allows the adjusting shims to flick out of the buckets, with disastrous consequences. The only way around the problem, and it should only be a problem in competition engines, is to fit the lighter, conventional bucket tappet and

Figure 10.9 Bucket tappet clearance adjustment systems



shim. When this is done, special valve spring retainers will be required to keep the shim in position between the valve stem and tappet.

This problem of the adjusting shim jumping out of position can also occur with the conventional bucket tappet and shim, but it is very unusual. In this instance valve float allows the shim to move out of the valve spring retainer recess and become wedged between the top of the spring retainer and the tappet. Instead of the valve closing, it is held open maybe 0.200in, whereupon the piston bangs into the valve and proceeds to destroy the motor. The more usual problem with the conventional bucket and shim is with shims that have been ground with faces that are not perfectly parallel. When this happens the shim rotates in an eccentric path, which wears away the top of the valve spring retainer. As a consequence the retainer collapses and allows the valve to drop into the piston. To ensure that this does not happen, measure all adjustment shims of this type with a micrometer, to ensure that their faces are parallel within 0.0003in. Also make sure that the shims are of almost the same diameter as the recess in the valve spring retainer.

The conventional flat tappet used in nearly all push-rod-type engines is losing ground to the mushroom tappet in competition where roller cams are banned. Earlier we discussed the relationship between cam velocity (and eccentricity) and tappet diameter. We are able to increase the eccentricity of a cam by increasing the tappet diameter. Unfortunately, most blocks cannot be enlarged to accept larger-diameter tappets, so the next best thing is to make the face of the tappet larger.

Enlarging the tappet diameter means that we can increase cam velocity, which results in a gain of around 15% in area under the displacement curve while retaining the same cam duration and lift. This means that we can improve performance due to the 15% flow improvement without the disadvantage of a loss of mid-range power if a hotter, long-duration, long overlap cam were fitted to give us the same 15% increase. From Table 10.6 you will see what this means in actual performance. The motor in this instance is a 350 Chevy set up with good mid range power in mind for road circuit racing. Each cam was chosen to give the best compromise between maximum power and mid-range power. The inlet rocker arms were changed to a 1.6:1 ratio to increase inlet valve acceleration. At 7,000rpm you will note that there is very little difference in maximum power between the three cams. However, down at 5,000rpm and below you can see the power margin widening. This is due to the difference in valve opening rates of each type of cam. Of course, if we had wanted more top-end power, while retaining the same bottom-end power as with the flat tappet cam, we could have achieved this too with the mushroom tappet and roller tappet cams. Many cam grinders are producing high-velocity cams for use with mushroom tappets, so take a good look at this performance route before resorting to the more expensive roller tappet set-up.

Overhead cam engines utilising bucket tappets can also gain a similar performance boost even when it may not be physically possible, or perhaps desirable, to increase the tappet diameter. In this situation we have the option of either going with an oval bucket design, or we can use a round bucket with a ramped face. For example, an oval bucket with a lobe contact face 30mm across by 35mm long replacing a round bucket tappet 30mm in diameter will be lighter than a round 35mm bucket, but offers the same increase – about 16% – in maximum safe cam and tappet velocity as the larger 35mm bucket. A round bucket tappet with a ramped face, to keep

Table 10.6 Engine dynamometer results with a 350 Chevy using flat tappet, mushroom tappet and roller tappet camshafts

Test 1 Flat tappet 0.842in diameter

rpm	*hp*	*torque (lbf ft)*
4,500	346	404
5,000	401	421
5,500	462	441
6,000	506	443
6,500	541	437
7,000	569	427
7,500	567	397

Test 2 Mushroom tappet 0.960in diameter

rpm	*hp*	*torque (lbf ft)*
4,500	362	423
5,000	415	436
5,500	471	450
6,000	515	451
6,500	547	442
7,000	572	429
7,500	570	399

Test 3 Roller tappet

rpm	*hp*	*torque (lbf ft)*
4,500	382	446
5,000	430	452
5,500	479	457
6,000	525	460
6,500	556	449
7,000	574	431
7,500	569	398

the edge of the tappet from being struck by the cam lobe, offers the potential for even higher cam and tappet velocity rates. Alternatively if minimising valve train weight is of greater importance than being able to run a more aggressive cam lobe then the [illegible] diameter can be dramatically reduced without any need to resort to a milder lobe profile. Porsche in fact went this route with their 3.6ltr GT3 24-hour endurance [illegible]

Apart from higher cost the downside with both oval and ramped face designs is that neither type is able to rotate (the ramped bucket is keyed to prevent rotation), so lobe and tappet face wear rates are higher. However, with diamond like coatings (DLC) applied to cam lobes and tappets this need not be an issue.

If your sport requires the use of more than 8,500rpm from a push-rod type engine, or super low end power, roller tappets are the trick set-up. More radical cams can be used since the cam velocity can be increased past that possible with flat or

The rev kit preloads the roller tappet against the cam lobe during the period when the ... is seated

mushroom tappets. Because the roller, by its very nature, rolls over the cam, lobe scuffing is virtually eliminated, which leads to longer cam and tappet life. To gain full benefit from the roller tappet and cam's rev potential (10,000rpm for the small-block Chevy V8), greater valve spring pressures are required, so take care to follow the cam manufacturer's recommendations carefully. A 'rev kit' will be required to provide the ... per tappet preload for high-rpm operation. The rev kit consists of an extra set of springs that fit between the tappet and the head, which load the tappet during the valve-closed period and keep it in contact with the cam. Additionally in a heavy duty or endurance race environment the rev kit functions as an engine saver by preventing a tappet coming out of the block in the event of a valve spring, rocker arm, or push rod failure. With all the lifters held in place in the block engine oil pressure is maintained which may enable the engine to finish the race down one cylinder, or if that isn't possible at least prevent further engine dam...

PUSH RODS AND GUIDE PLATES

Two requirements for an e... a high performance ... weight and rigidity, light weight to allow an increase in operating rpm, and rigidity to ensure accurate valve operation to attain the desired rpm. Push rods and rockers require particular attention with regard to both of these important but conflicting ideals. Where possible, I always replace push rods with those made of aircraft grade seamless 4130 chrome-moly steel tubing, heat treated to 120,000psi tensile strength. These must be of the correct length to maintain correct rocker arm geometry.

Basically, rocker arm geometry is correct when the centre of the rocker ... coincides exactly with the centre-line of the valve stem with the valve lifted 40-5... its total lift. If this is not the case, the valve guide will wear to an oval shape very ... due to the increased side thrust, which ... valve seating and ... problems. Rocker arm geometry may b... by the installation ... cam, or high-ratio rocker arms, but may also be changed by head and block deck machining...

Also to be considered at this point are push rod guide plates. If the standard plates have a history of breakage, replace them with plates of a better design or made of better material. Any motor that has ball-fulcrum-type rockers must be fitted with guide plates if reliability is to be maintained. As well as allowing up-and-down

action, these rockers unfortunately engage in a side-to-side motion, which does a great job of wrecking the valve guides. The only way to curtail this sideways twist is to install push rod guide plates. Generally, I have found that standard motors not fitted with guide plates suffer twice as much valve guide wear in line with the cylinder head (180°) rather than across the head. It can be imagined how much more the rocker is going to be pushed sideways when stiffer valve springs are fitted and cam acceleration rate increased, when there are already problems with the standard arrangement. Remember when you fit guide plates that hardened push rods will have to be fitted to prevent rapid push rod wear.

THE ROCKER ASSEMBLY

Next in line for scrutiny are the rocker arms. These transmit the tappet motion to the valve and may be pressed steel or forged items. Generally, stock forged rockers are reasonably rigid and will do a good job in a high-performance engine, with some modification. The rocker should be lightened with a view to retaining the vertical section while reducing the horizontal section. The rocker does most of its work directly in line with the push rod and valve stem centres, so in most instances material can be removed without weakening the component. The area over the valve stem must be left wide enough to give full valve stem contact. The shank of the rocker should be left alone except to remove 'dingle berries'. Shot peening is to be recommended in racing applications. Obviously there is no point in removing material close to the rocker shaft as this mass is of no consequence.

Most forged rockers are supported by a light steel rocker shaft and posts of steel or light alloy. The Austin Mini-type shaft and posts are about as rigid as possible in production form. However, the same cannot be said of the four-cylinder British Fords which leave a lot to be desired in this department. If any modification is intended for one of these, no matter how mild, the alloy rocker posts must be discarded and replaced by more rigid steel posts. The use of a moderately hot cam will also make it

A shaft mounted rocker assembly provides a much better base for reliable valve operation at high rpm with heavy valve spring pressures.

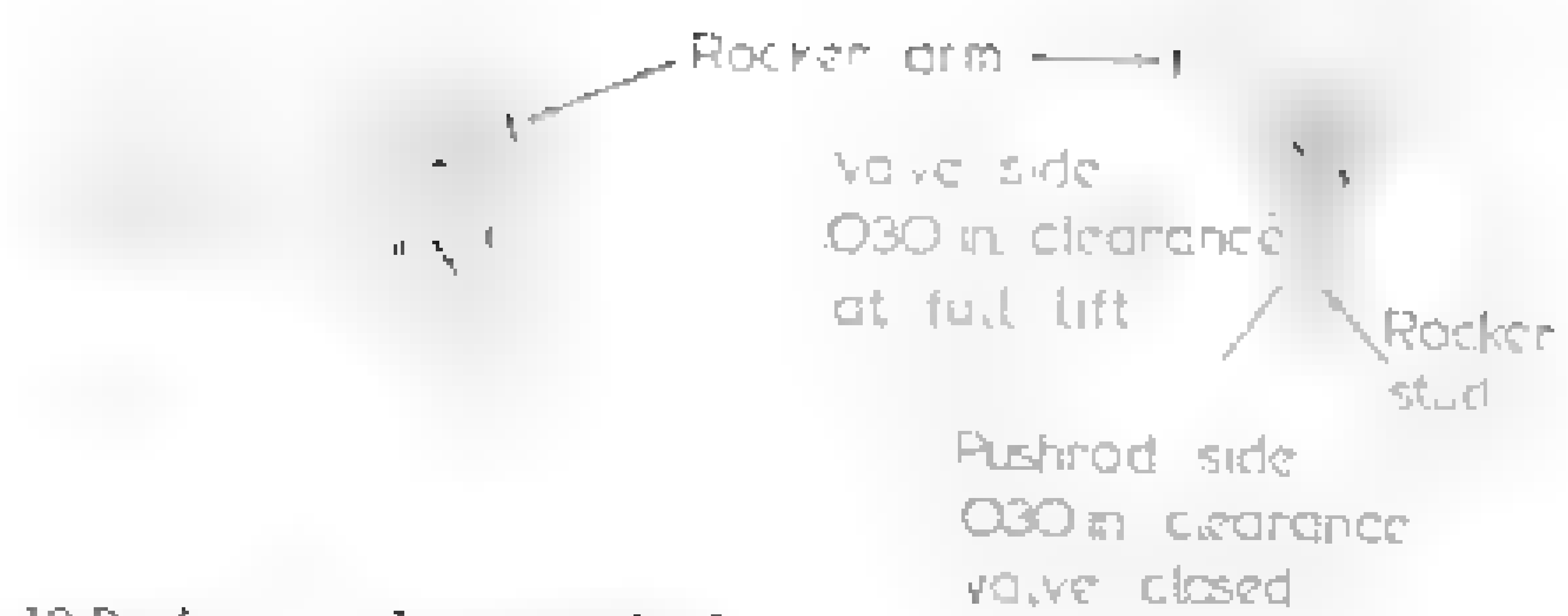


Figure 10.10 Rocker arm clearance check

necessary to use a heavy duty steel rocker shaft. The ultimate is without doubt the Piper integral rocker box, which supports the outer ends of the shaft very effectively

The separation springs fitted between the rockers should be replaced by tubular sleeves made of steel. While doing this, pay attention to fitting sleeves of the correct length to centralise the rockers over the valves

Pressed steel rocker arms are another story. While they have been used with some fairly warm cams, I do not like them, and I would not use them in any motor fitted with anything wilder than a sports cam. The alternative is expensive but well worth it. These are made of aircraft-quality, extruded high-density aluminium or forged steel, and are fitted with needle roller bearings and a roller tip. Assuming that your rocker geometry is correct, these roller rockers will reduce valve guide wear by 'rolling' over the valve stem as it pushes the valve open. Standard rockers scuff the valve to the side, which wears the stem and valve guide. Generally, I have found that roller rockers reduce the oil temperature of Chevys by up to 20°F, due to reduced friction, which means that we are picking up power as well

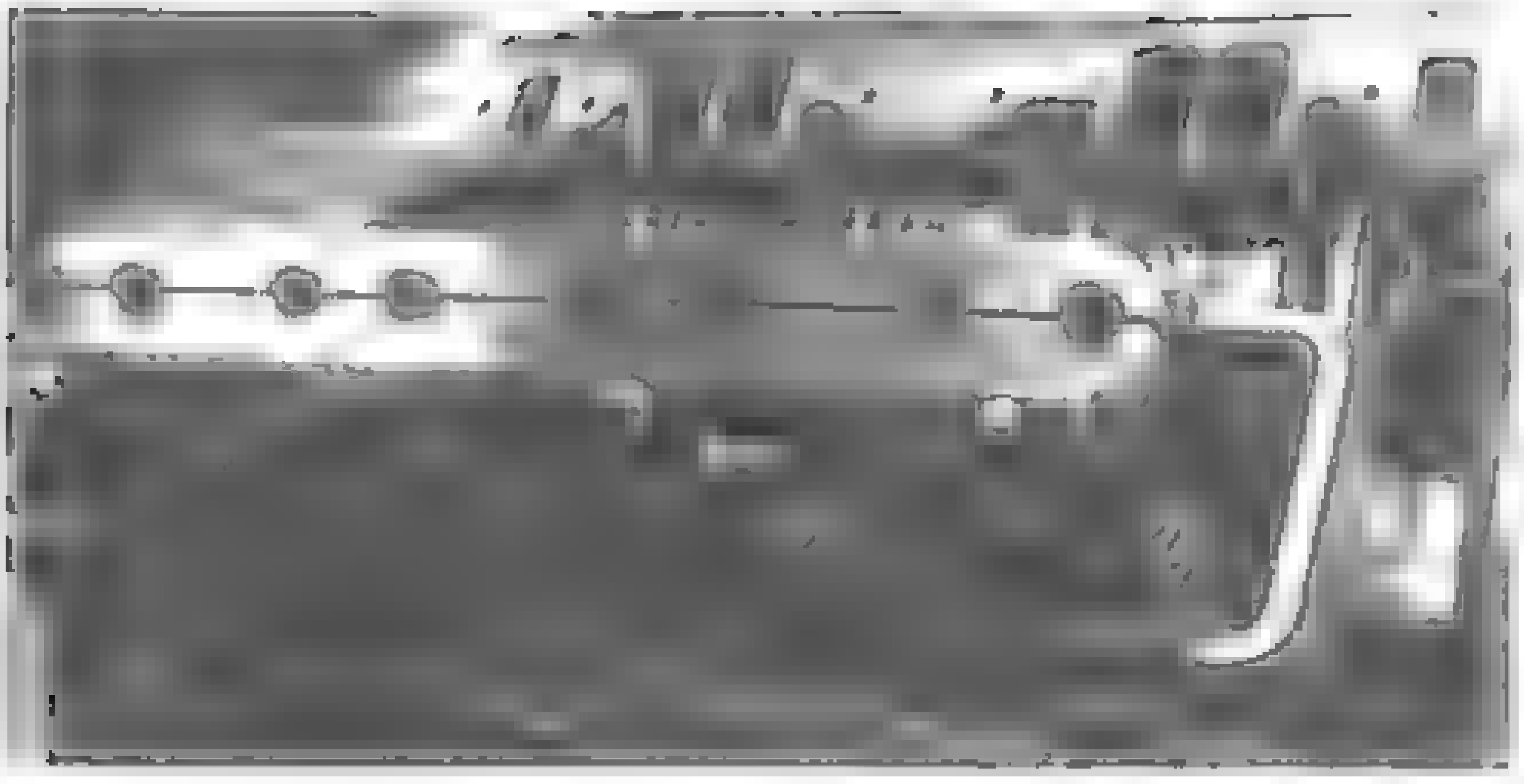
Ball-type rockers require attention when high-lift cams are installed, to ensure adequate working clearance between the end of the slot in the rocker arm and the rocker stud. This clearance should be measured using a wire 0.030in thick (a paper clip is fine), with the valve closed and also at full lift (Figure 10.10)

Table 10.7 Replacement rocker arms

Engine	Standard ratio	Alternative ratio
American Motors 290–401	1.6:1	
Austin 'A' series	1.25:1	
Buick V6		
Chevy small-block and V6		
Chevy big-block		
Ford Cleveland		
Ford Windsor		
Ford V6 (British)		
Ford big-block 332–428		
VW 1200–1500	1.23:1	

*Requires offset studs

**Shaft mounted rocker arms



When budget constraints put a shaft mount rocker arrangement out of reach the next best path is heavy rocker studs and a rocker stud girdle to spread the load

An absolute must is to replace pressed-in rocker studs with screw-in studs, if a race cam is to be fitted. Many push-in studs are marginally stable even with the standard cam and valve springs (eg English V6 Capri), so if the standard cam is being retained or if a sports cam is being fitted, at the very least the push in studs should be pinned. When screw-in studs are being fitted, seal them with gasket sealant to prevent water seepage if they break into the water jacket.

If possible, 3/8in rocker studs should be replaced with 7/16in studs, to improve rigidity. A rocker stud girdle is also a good investment as these brace all the rocker studs against each other. This reduces the load on each stud and consequently the amount of flex, as all studs share the load when a valve is opened. With canted valve heads it is not always possible to brace all the studs together using a single girdle, but generally two studs can be girdled together for load sharing. Some girdles use a single brace bar with 'U' bolts clamping the bar to each stud. This type does a reasonable job but I prefer a double brace bar girdle for added rigidity. With proper setting up, it is possible to [illegible] push-rod-type valve gear almost as rigid as single ohc valve gear employing rockers.

Generally, rocker arms have a ratio of 1.2–1.7:1. This means that rocker arms with a 1.5:1 ratio will lift the valve a theoretical 0.600in if the cam lift is 0.400in. Many racers get carried away by fitting rocker arms of a higher ratio than standard to increase maximum valve lift. This is not the purpose of the exercise at all. What we are after is to lift the valve off its seat faster in the initial stages of opening, to improve mixture flow and cylinder filling. The a[illegible] increase in maximum lift is of little consequence except to cause us more problems with valve spring coil bind, valve t[illegible] piston clearance, etc.

High-ratio arms are different from their standard counterparts by having a shorter distance from the push rod centre to the rocker stud (or shaft). The distance between the rocker stud (or shaft) and the valve stem is fixed (unless the rocker studs are re-positioned or offset studs are fitted), so the only way that the rocker ratio can be increased is by decreasing the rocker arm length on the push rod side. For this re[illegible] possibly the guide slots in the push rod guide pla[illegible] and also the holes in the rev k[illegible] bar, will require elongation to prevent push rod fouli[illegible]

Before fitting high-ratio rocker arms, check with your cam grinder regarding the changes that will have to be made in the valve train. Remember that the velocity and acceleration characteristics of the cam are being changed when the rocker arm ratio is changed, so will the cam, tappets and valve springs be able to handle the extra forces being imposed on them? Too many tuners, and cam grinders as well for that matter, feel that you can just go ahead and use any rocker arm ratio you wish. If this were the case, life would indeed be much easier. Unfortunately, modern-day camshaft profiles are often not designed for a specific engine with a specific valve train, so rocker arm ratio can be varied without major problems, but check with the cam grinder first. If all cams were designed as an integral part of one specific motor, with a certain specific amount of valve train flex, as they should be, a change in rocker arm ratio could mean something as drastic as a cam change as well!

Think carefully before changing standard rockers for high-ratio rockers. If you do not need more low-end and mid range power, you will be wasting your time fitting them. Most performance cams give ample total valve lift without having to worry about fiddling with rocker ratios. If your valve train is already marginal in the reliability department, a change to high-ratio rockers will worsen the problem, so be careful.

VALVE SPRINGS

The valve springs have the unenviable task of keeping the whole act together and functioning smoothly. Therefore the cam grinder should be able to give you all the information you need to make that camshaft operate as designed in your engine. This must, of course, include the following vital details: recommended valve spring, installed height and on-seat pressure, and full lift pressure. Also a recommended safe rpm limit should be stated for a valve train correctly set-up.

Valve spring selection is especially critical in engines with aggressive cam lobe shapes, heavy valves and valve train components, running at high rpm for long periods. This basically means all large push rod race engines and endurance race engines without air restrictors. Buy the best quality springs you can find. They will be expensive because the purest, cleanest steel such as is produced by Kobe Steel (Kobe, Japan), and special drawing dies and winding techniques, along with precise heat treatments and shot peening, and many inspections during each phase of manufacture, doesn't come cheap. I prefer to source springs from suppliers with a consistently good product like Schmitthelm or Kurt Kauffmann, or in more recent times also Endura-Tech.

Regardless of where you source your springs, if you are competing in a tough environment such as described above, be prepared to devote a lot of time to carefully sorting through each new batch, weeding out those that aren't up to spec, and then properly finishing those that pass. First off inspect the wire for nicks, scratches, indentations, or deformity under a 4X magnifier. Next measure the height and spring rate. Then check for squareness by determining that the spring stands straight when placed on a flat plate – check this from both ends. All springs that pass these inspections should now be polished on both ends. You don't want any sharp edges anywhere on the spring that could lead to breakage, and you don't want any sharp ends nicking or wearing into the next coil as the spring compresses. Likewise you don't want sharp edges cutting into the spring retainers or spring seat.

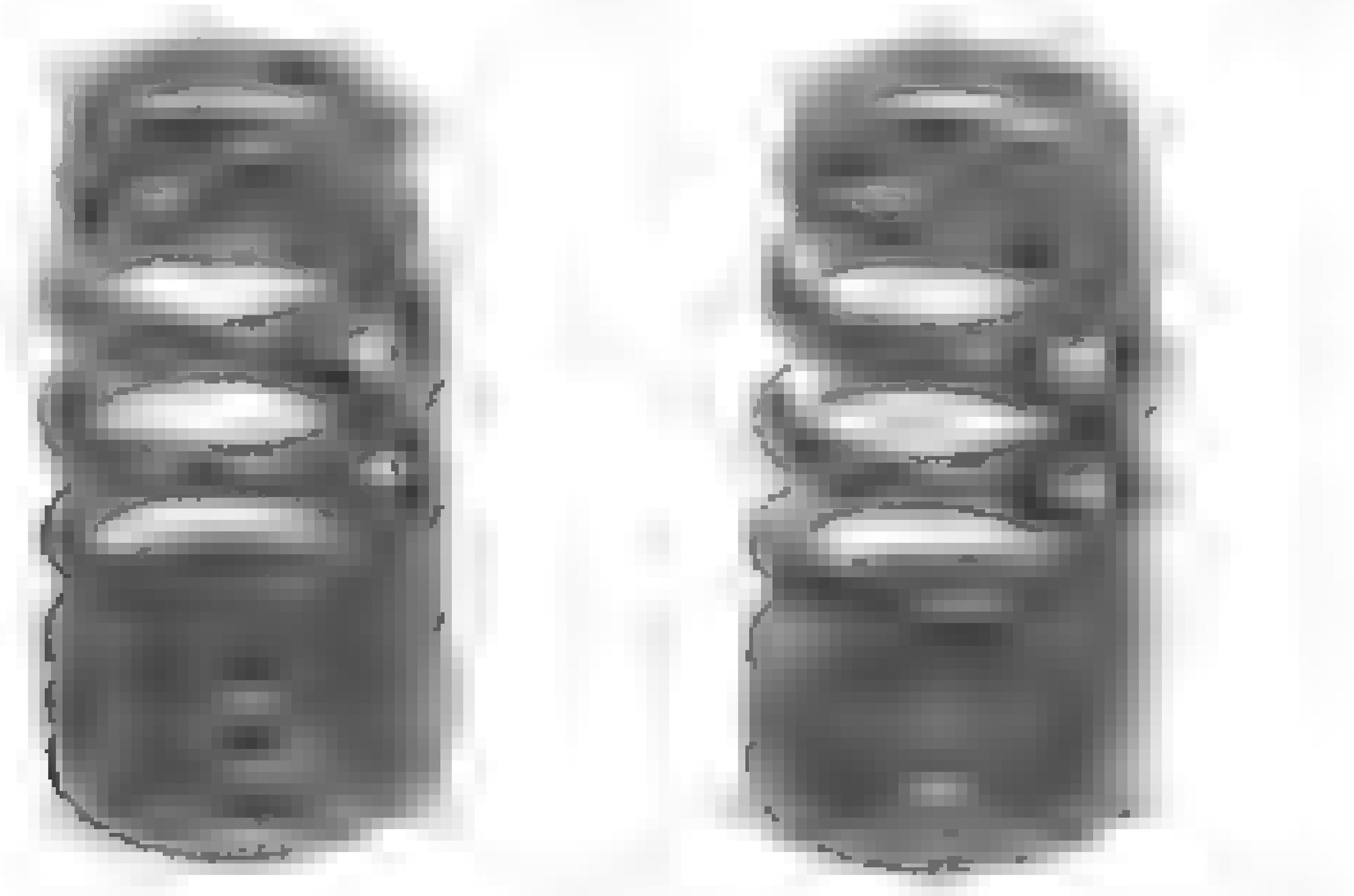
Valve springs have to be heavy enough to keep the tappet in constant contact with the cam lobe and allow a possible over rev from a missed gear change, yet also be light enough not to cause excessive cam and lifter wear and valve train flex, or rob us of power.

Valve spring surge can lead to erratic valve spring behaviour at certain rpm, caused by vibrations set up in the valve train. These vibrations excite the natural frequency characteristics of the spring, and when this happens a surge reduces the available valve spring tension, its force opposing the valve spring tension. This can lead to the tappet losing contact with the cam lobe, the possible end result being valve float, valve to piston contact and increased valve train wear.

To help overcome the problem, spring manufacturers have used several approaches to dampen surge. One takes the form of a damper coil of flat spring steel, which dampens by friction through contact with the inside of the coil spring. Another is the use of counter-wound inner springs, with coils wound in the opposite direction from the outer springs. Variable-pitch coil springs with a progressively smaller spacing between coils towards the bottom of the spring have also proved successful. Possibly the most common approach, in production engines at any rate, is to close up completely the coils at the bottom of the spring, and also sometimes at the top.

A more recent approach has been the adoption of beehive-shape springs in some applications that can get by with a single spring. GM used this type in the 5.7ltr V8 Chev LS1 released in 1996 and it is being used in NASCAR restrictor plate engines. The design has a number of advantages over the conventional shape valve spring. First, its reduced diameter at the top allows the use of a smaller and thus lighter, by about 5gm, valve spring retainer. Secondly, because this reduced diameter at the top cuts the amount of leverage trying to bend the valve stem, valves with a thinner stem to improve air flow and reduce valve train weight – or lighter valves with a hollow stem can be fitted. Thirdly, it allows a spring pressure reduction at full lift of around 10%–20% without any reduction in safe engine speed and an on seat pressure decrease of 15%–25%; cutting both parasitic losses due to friction, and valve seat wear. This is possible because of the beehive spring's superior harmonics. Each coil increases in diameter when compared with the coil above, so each coil has a different natural frequency which helps dampen spring surge.

Beehive-shaped valve springs help reduce spring harmonics and enable lighter springs to be used.



The natural frequency of the valve spring should be five times that of the engine rpm. Therefore if the engine speed is 7,500rpm the valve springs should have a natural frequency of around 37,500 cycles per minute. Interestingly, reducing the number of working coils (ie those not wound so close as to touch together) increases the natural frequency of the valve spring. This has another benefit to the racing spring manufacturer in that the reduction in working coils will also reduce the solid stack height of the spring. Therefore the spring can be compressed further before coil bind occurs.

Another development has been the use of ovate rather than round wire in the manufacture of springs. In cross-section such wire has a shape similar to a chicken egg. This means it has greater surface area, and thus greater strength than conventional round wire of the same thickness. Also the added surface area allows heat to be dissipated more easily, increasing spring durability and life. Round wire with the same spring characteristics would result in a valve spring that was heavier and taller. Then to stop coil bind a longer heavier valve may be required if the valve spring seats could not be cut deeper.

CHECKING FOR VALVE SPRING COIL BIND

Just how much clearance should there be between working coils at full valve lift? That depends on the spring design, but I always work to a minimum of 0.008in between working coils, although I like to aim for 0.010-0.012in. Some spring manufacturers specify figures as low as 0.005in and as high as 0.020in. If they specify a figure higher than my recommendation, follow their instructions. This safe working clearance between coils must always be maintained, otherwise the valve spring will stack solid with disastrous consequences.

Simply providing a safe working clearance between coils doesn't guarantee that the spring is performing at its best; nor does following the spring manufacturer's minimum installed height figure. What is the best installed height for that particular cam lobe, valve train and spring combination, from the aspects of spring life and engine performance, can only be assessed from extensive Spintron testing.

T[illegible] of an increase in valve lift, many heads will require modification to prevent coil-binding when a high-lift cam is installed. Others may not require the valve spring seat to be deepened but may require the valve guides to be slimmed down to permit the installation of inner springs. Either operation can be performed with a counterboring tool of the appropriate dimensions. To determine how much deeper the spring seats must be machined (if at all), it will be necessary to fit the valve and valve spring retainer and locks. Pull the retainer up tight and measure the distance between the spring seat and bottom face of the retainer. If the measurement is less than the valve-open or installed spring height specified by the cam grinder, it will be necessary to deepen the seat. This must be repeated for each valve, and repeated each time the valves are replaced or re-faced, or if the valve seats are re-cut.

If the valve springs are set up at the on seat height specified (and they should be otherwise your full lift spring pressures will be all over the place, possibly allowing valve float), you should not have any problems maintaining the correct clearance between coils. However, check the clearance at full lift, using a feeler gauge to be sure, and do not forget to check the inner springs.

Pneumatic valve control

On overhead cam motors using bucket-type cam followers it is not possible to check for coil bind by using a feeler gauge between the coils. What you must do in this instance is to count the number of working coils and multiply that by the required clearance between each coil. If this works out to be, say, 0.036in, insert a feeler gau

of this thickness between the cam lobe and follower and turn the cam over to full lift. The cam may lock up before full lift, which would indicate insufficient working clearance. To rectify this, the valve spring seat will have to be machined deeper. Just how much the seat will have to be machined can be found by inserting progressively thinner feeler blades until the cam will turn freely. If the cam turns with a 0.023 blade inserted, this means that the spring seat must be deepened by 0.036–0.023in = 0.013in. This procedure should be repeated for each valve spring.

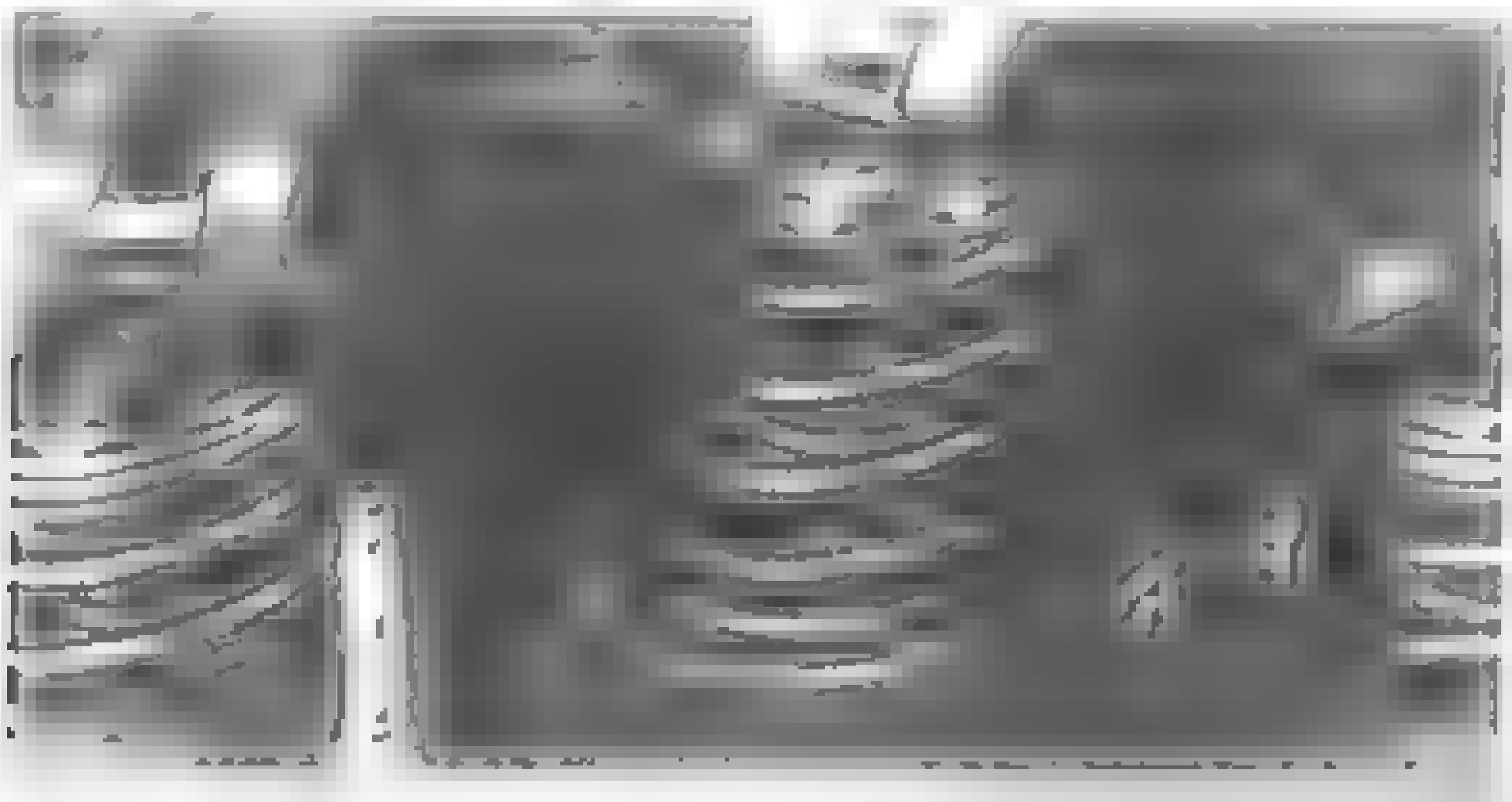
While we are on the subject of bucket tappets, there is another very important clearance check that must be made. When high-lift cams are installed, it is possible for the tappet to run out of travel and bang into the head. To check for sufficient travel it will be necessary to fit just the cams and tappets to the head (leaving out the valves and springs). Turn the cam over by hand until you have a lobe in the full lift position and see if you can fit 0.060in of feeler strips between the nose of the cam lobe and the face of the cam follower; carry out this check on every tappet. If you can fit only 0.052in of feeler strips, you will need to remove 0.008in from the lower edge of the tappet; up to 0.015in can be removed, but if that is insufficient it is better to machine the head. Put a nice radius on the lower edge of the bucket after any machining is done.

VALVE SPRING PRESSURES

Remember that valve springs wear out. High rpm and excessive heat accelerate this natural fatigue process. We can't do much about high rpm, but we can spray the springs with a constant flow of oil to transport some heat away. On a competition motor this aging process may cause the spring to break or it may lose tension and become weaker. If you are a racer, check the spring tension each time the head is removed. The pressure at the on-seat and full-open lengths should check out within a few pounds of new springs. If any are not up to specification they should all be replaced.

Many people are rather mystified by any reference to 'spring rate', so it might help to explain just what it is and how we work it out. Firstly, spring rate is defined as

Valve spring cooling is critical in endurance race engines and those operating big valves at high rpm. The simple solution is to locate an oil spray bar in the rocker cover to spray each spring, but this path adds weight and complexity with external oil pipes to each rocker cover. A better approach is to have an oil gallery in the rocker bed with a series of screw-in jets to meter oil onto each spring.



the amount of build-up in pressure when the spring is compressed. For example, if we had a spring with an on-seat pressure of 100lbs @ 1.625in, and a full lift pressure of 300lbs @ 1.125in, it would have a spring rate of 400lb per in, which would mean that [illegible] compressing the spring 0.5in (1.625 to 1.125in) resulted in the spring pressure [illegible] spring 1 inch would require a load of 400lb.

This should help you to appreciate the need to set up valve springs at the precise specified height. If the springs in the case quoted were set too high, the on-seat [illegible] with correctly installed springs was 7,500rpm, this would reduce it to 7,115rpm.

$$\sqrt{\frac{270}{[illegible]}} \times 7{,}500 = 7{,}115\text{rpm}$$

When the on-seat height is more than specified, shims of the appropriate thickness [illegible] A superior method of achieving the required spring height is to use collets with [illegible] then be raised or lowered by fitting the appropriate collets.

Sometimes it is possible, and desirable, to increase the rpm at which valve float will occur. Providing that the valve train and camshaft are capable of running a higher speed (check with your cam grinder), stronger valve springs will have to be installed. To find the required full lift pressure of the new springs, use the following formula:

$$NP = \frac{Nrpm^2 \times P}{rpm}$$

where NP = the new valve spring pressure required, P = the present valve spring pressure at full lift, Nrpm = new valve float rpm, and rpm = the present valve float rpm.

A decrease in effective valve train weight will also increase the valve float rpm limit. [illegible] tappet lightening, and by fitting light rocker arms or lightening the original rocker arms. Remember, however, that a decrease in rocker arm weight close to the pivot point does not effectively decrease the valve train weight. The intake and exhaust [illegible] titanium-aluminium-vanadium alloy allows us to lose 35% of the valve's weight. A cheaper alternative for exhaust valves is to use hollow-stem austenitic stainless steel stems, which are around 20% lighter than their solid stem counterparts. An effective decrease of 10% would increase the valve float rpm by the factor

$$\sqrt{\frac{100}{90}} = 1.054$$

Therefore if the old valve float speed was 7,500rpm, the new speed will be 7,500 x 1.054 = 7,900.

Figure 10.11 Bad valve and collet designs.

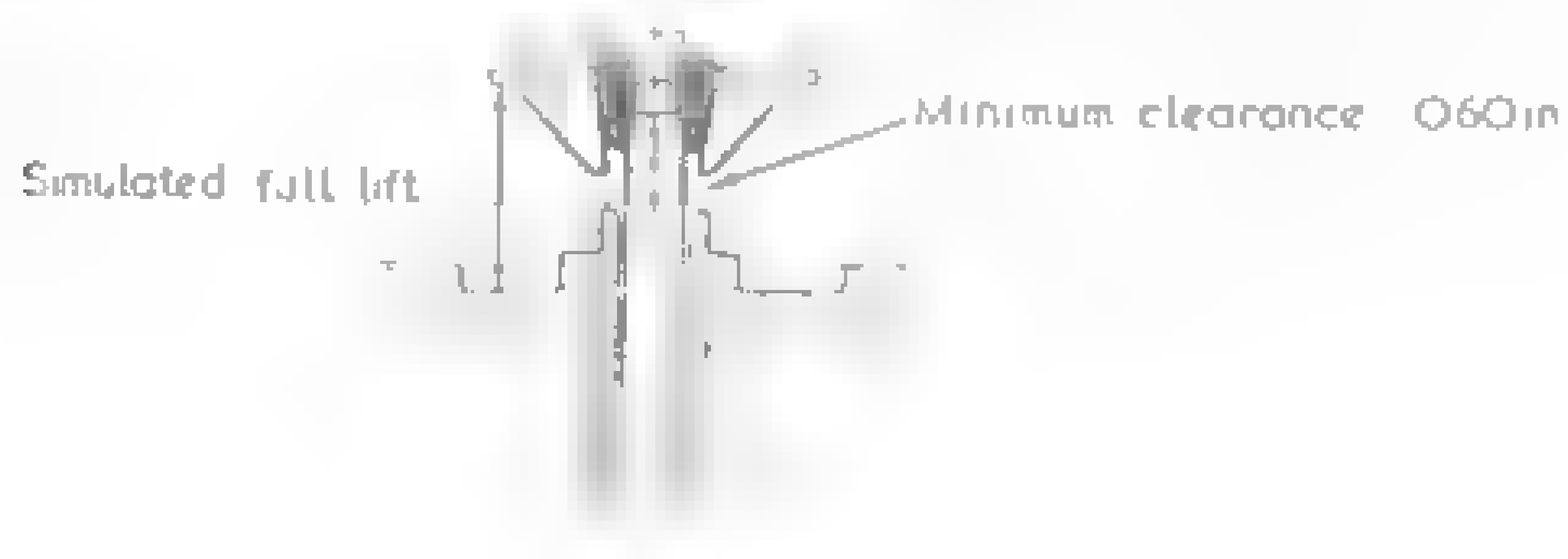
VALVE SPRING RETAINERS

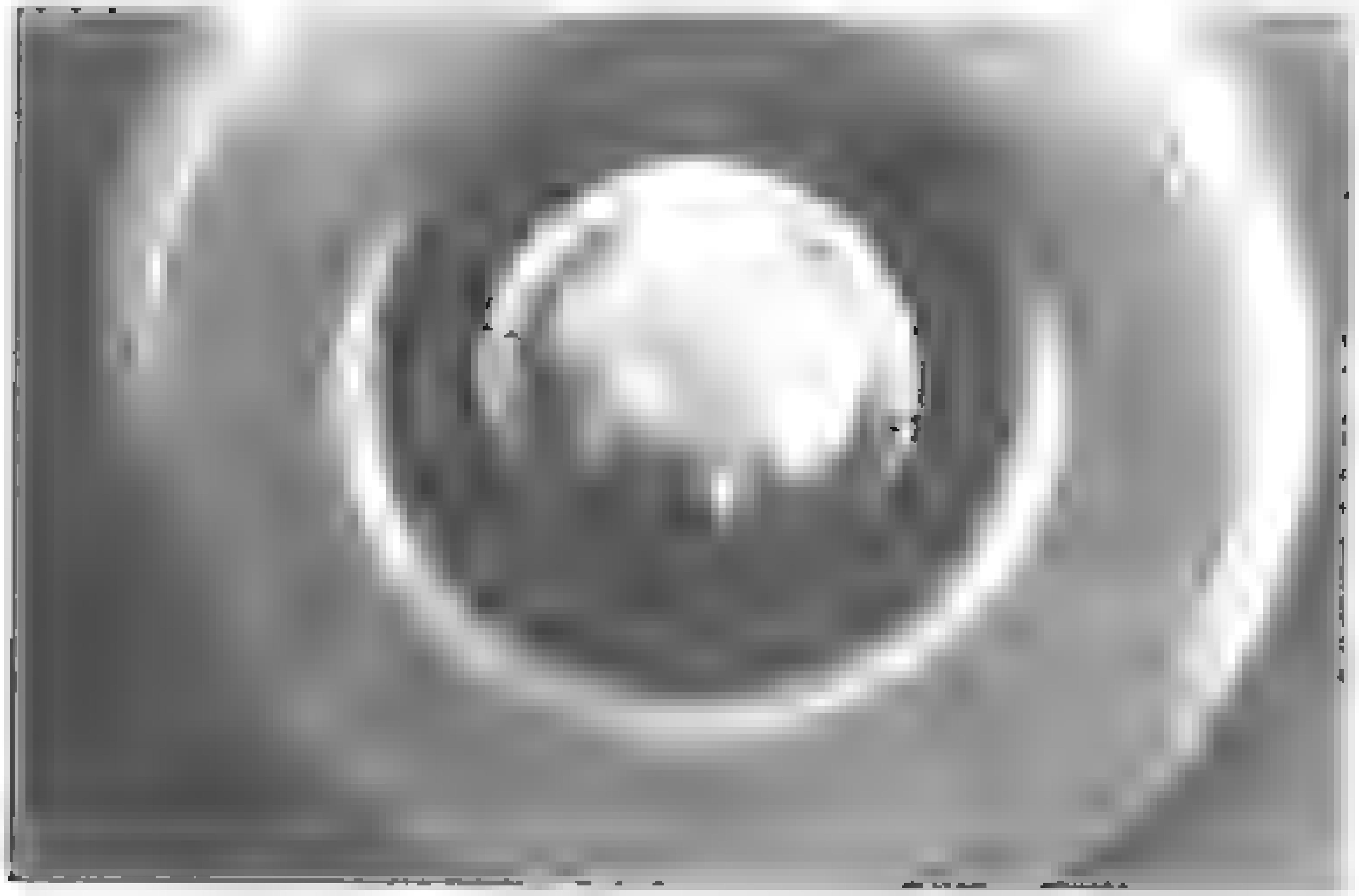
Valve spring retainers and collets (or valve stem locks) require attention to maintain [illegible] Austin engines (Figure 10.11). This design is very bad, as the stress concentration at the square shoulder of the collet groove is sure to cause failure. Another problem design is that where the two collets actually butt together and allow the valve stem to lie loosely and float in the centre. This is done to allow the valve more freedom to rotate and clean the valve seat. What has to be done is to grind off the faces of the collets so that they clamp tightly on to the valve stem. When fitted, there must be a gap between both halves of the collet for this to be possible.

Standard spring retainers and stem locks will do a good job, but for ultimate reliability and a saving in weight, special pieces should be fitted. Titanium alloy retainers are the lightest/strongest available, while steel retainers of heat-treated 4140 bar stock are the next best, but heavier. I do not like aluminium retainers, and consequently do not recommend their use except in relatively unstressed conditions.

The next check that must be made is for interference between the bottom of the valve spring retainer and the top of the valve guide, in the simulated full valve lift condition. There must be at least 0.060in clearance at full lift, but I always aim for 0.100in. This clearance check can be done very easily without the heads being fitted to the motor. All that must be done is to set up a telescopic gauge or a snap gauge at the full open valve spring height, then fit the retainers to each valve and check out the

Figure 10.12 High lift cams can cause the valve spring retainer to hit the valve guide at full lift. The minimum clearance required is 0.060in.





Dropped valves can result when valve stem locks butt together like this

clearance between the retainer and valve guide when the snap gauge is fitted in between the retainer and valve spring seat. If there is less than 0.060in cle[illegible] metal will have to be machined from the valve guide. (Figure 10.12)

CAM DRIVE ARRANGEMENTS

Returning to the camshaft, we require some accurate and reliable means of driving it. Several methods are in use, but none are without their problems. For many years cams have been driven by single-row chains. These have proved themselves reliable in very high-speed racing engines, with open-valve spring pressures of 250lb, but chain and sprocket wear is a problem at high engine speeds. Double-row chains are more up to the job and can successfully handle open-valve spring pressures of around 420lb in excess of 8,000rpm. Note that we are talking about q[illegible] roller chains and steel sprockets, not the silent Morse chain running on fibre or nylon tooth sprockets. With a new chain and sprockets installed, the valve gear will normally retard the cam by [illegible]1°, and as wear takes its toll this retarding will increase. Therefore chain drive ca[illegible] should be advanced in the initial set-up to compensate for this.

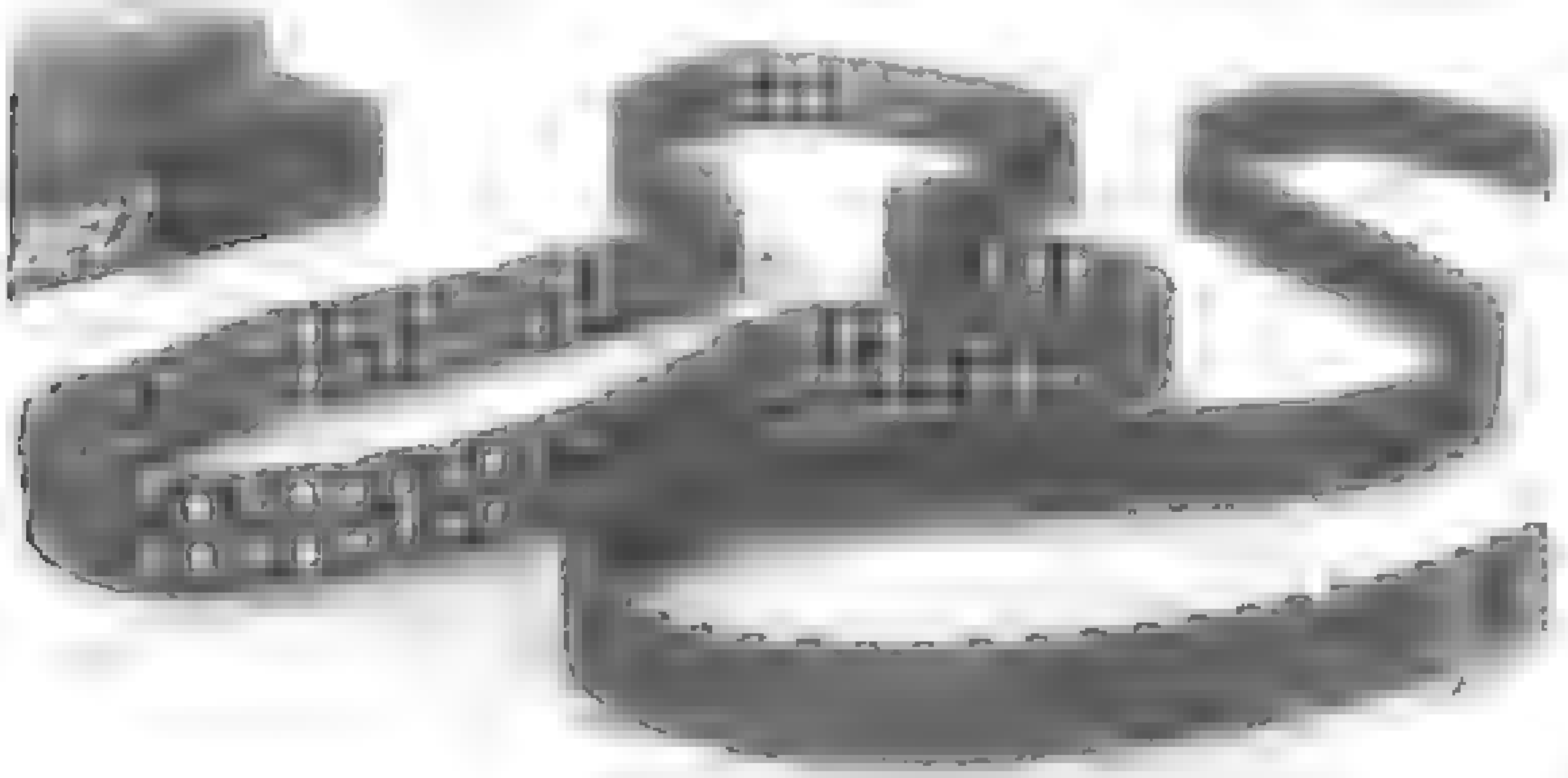
The toothed rubber belt also seems to be up to the task, having been Renault's choice for their 1.5ltr turbo Formula 1 engine running in excess of 11,000rpm almost 30 years ago. In more recent times many push rod American V8s in competition have converted to belt drive to provide more accurate valve and spark timing as a belt more effectively isolates crankshaft harmonics from the camshaft. However toothed belts are not completely problem free. First you need to be careful never to turn the engine over backwards or the belt may slip, altering the valve and maybe also the ignition

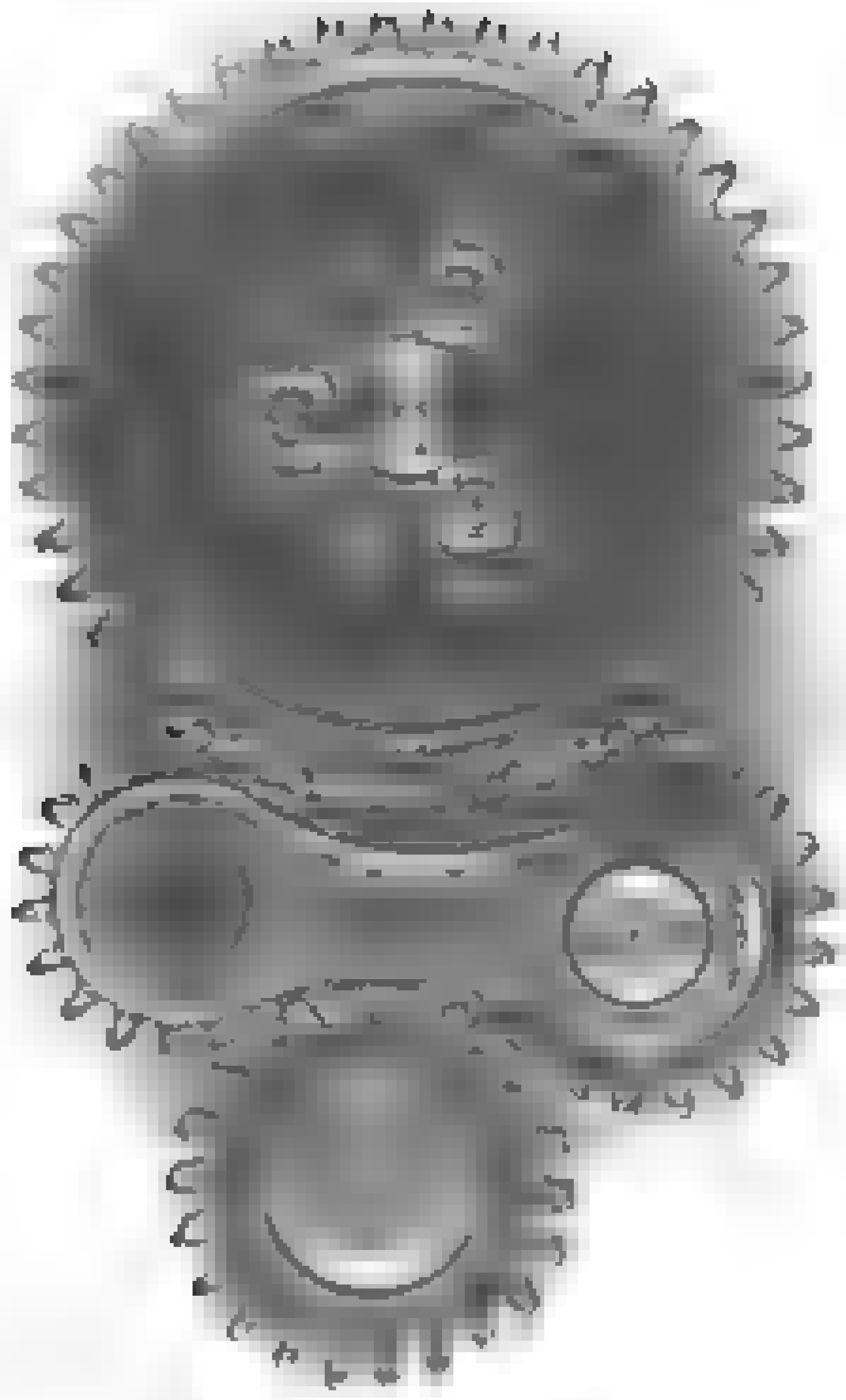
ming. Worse, the valve timing can be so far off as to cause valve to piston contact ... lso valve to valve contact in twin cam engines. Also when belts fail bad thing ... ppen inside the engine. With mild cam profiles the manufacturer's stock belt and normal replacement interval usually can be followed without problems. However, it is ... lways smart to check with your cam grinder to be sure, and be careful to use a good quality belt from a reputable manufacturer like Gates. If a heavy duty belt is available it is wise to fit that. Remember too that the factory gets it wrong at times. Lancia recommended belt replacement on the Intergrale four-valve at 100,000km intervals but many broke belts at only half that distance with standard cams – wrecking the head ... valves, and pistons too at times. When fitting the belt be careful not to kink it, and if reusing the belt ensure that it will rotate in the same direction as when first fitted. To avoid this problem I always fit belts in such a way that when facing cylinder No 1 a... lettering on the belt is correctly orientated. Stretch will also allow cam retard... with this type of drive, so allowances must be made when setting up the cam.

... some ways gear drive is the ideal system, but there are many complications. Align-boring (or tunnel-boring) the block will allow the crank gear to mesh tighter ... system to adjust backlash and meshing depth must be provided. If you had the job of setting the gear mesh on a Formula 1 engine you certainly would not consider ... ar drive to be the ideal set-up. If you change from chain drive to gear drive, and use the simple two-gear system, it will be necessary to install a reverse-ground cam as it will now be turning backwards. A reverse distributor gear will also be required, to keep that component turning the correct way. However, in spite of all this, gear drive is the best set-up for a competition motor.

Many motors use gears in standard production form, but generally these are useless, and are not capable of reliable service, even if only a sports cam is fitted. Production-type gears are designed with silent operation in mind, so are made of such material as compressed fibre, or nylon-coated steel and aluminium. If you intend to

A toothed timing belt can provide reliable service only if handled and fitted properl... and is replaced at the intervals specified by the cam manufacturer.





Steel or alloy gears provide reliable drive to the cam

extract any type of performance at all out of an engine with gear materials such as these you are in for trouble. The only way around the problem is to use quality steel gears.

Remember accurate ignition timing and performance also has its beginning with a good cam drive arrangement that will keep the spark in step with piston motion. This is one reason why some ignition systems now work directly off the crankshaft and others use sensor-trigger magnets attached to the flywheel or harmonic balancer. As well as driving the distributor or magneto, the cam may also look after the chores of supplying drive for the oil pump and fuel pump, as well as mechanical fuel injection, if fitted.

CAM FITTING AND RUNNING-IN

Now that we have covered the theory of the camshaft and valve train, let us get down to the actual camshaft installation. At this stage we will assume that you have already read the chapter on block preparation and the cam is ready to drop in (if the motor has an overhead cam the same information covered in block preparation will apply, but in this instance to the head). You will note that the cam possibly has a black coating; this should not be removed except from the bearing journals. Freshly ground steel does not have any oil retention ability, and as lubrication of the high-performance cam is of the utmost importance during the first few minutes of engine operation when running in the cam, we give the cam a special treatment and coating. I also treat the tappet faces in the same way. This treatment is called Lubrite, Parkerising or Parko-Lubrite. Actually the last name best describes the treatment, albeit the reverse of the way in which we actually apply it. The cam bearing jou[illegible] are taped and the cam is phos[illegible]

into a high temperature bath of phosphoric acid, which etches open the pores of the cam so that it may retain oil more effectively. This is the Lubriting part of the process. Next the cam receives a phosphate coating to aid the breaking-in process.

To assist lubrication even further, always coat the cams and lifters with a mixture of high-pressure Hypoid 90/140 gear oil and molydisulphide; alternatively, if the engine will be running on castor oil I use Castrol R40, as castor oil must not be contaminated by mineral or synthetic oils. The first 10–15 minutes is the critical running-in period, and during this time engine speed should be maintained at [illegible] notably the push rod and ohc Cortina/Pinto, require special care, and in their case [illegible] for 40–45 minutes. After this, the racing cam should be given more treatment, a 4,000–5,000rpm run for 10–15 minutes, but taking care not to let the engine overheat.

[illegible] measure the end float. This should be 0.002in, more than 0.005in is unacceptable. If we do not control 'cam walk' within fine limits, we can upset the whole valve train. Remember that flat tappet cams have the lobes ground at an angle, which tends to push (or thrust) the camshaft as the lobes open and close the valves. This thrusting backwards and forwards also has a bad influence on the ignition timing, as the distributor is usually driven off the camshaft. When mushroom tappets are used, there is the possible problem of the tappets overlapping two cam lobes if cam walk is not precisely controlled.

SPLIT, ADVANCED AND RETARDED TIMING

The next thing we must do is to time the cam so that the opening and closing of the valves is in a special 'tuned' relationship with the up-and-down motion of the pistons. Unfortunately, many feel it is sufficient to line up the timing marks and leave the valve timing at that. With a sports cam that may be so, but you will not be able to achieve optimum performance. Due to manufacturing tolerances the cam may be several degrees advanced or retarded. This may be the result of timing marks, keyways or dowels being out, or in ohc designs could be due to surface grinding of either the head or block. Additionally the cam lobes can also be out in relation to the cam dowel or keyway due to grinding error.

Something else that must be taken into consideration is that even if the cam timing was spot-on according to the manufacturer's figures just by lining up the timing marks, would that necessarily be the best position for the cam for optimum overall performance? Figure 10.13 illustrates three timing diagrams that represent one complete cycle (two revolutions, or 720°) of the engine. The first diagram indicates that the cam has symmetric (or split) timing of 30°–70°/70°–30°. You will also note that the inlet valve is fully open (maximum lift) at 110° after TDC, and that the exhaust valve is fully open 110° before TDC. The second diagram shows that the cam has been advanced 6° from the manufacturer's recommendation to produce a timing of 36°–64°/76°–24°, with full lift occurring at the inlet 104° after TDC and at the exhaust 116° before TDC. In the third diagram the cam has been retarded 4°, with a timing of 26°–74°/66°–34°. Maximum lift will now be at 114° after TDC at the inlet and 106° before TDC at the exhaust valve.

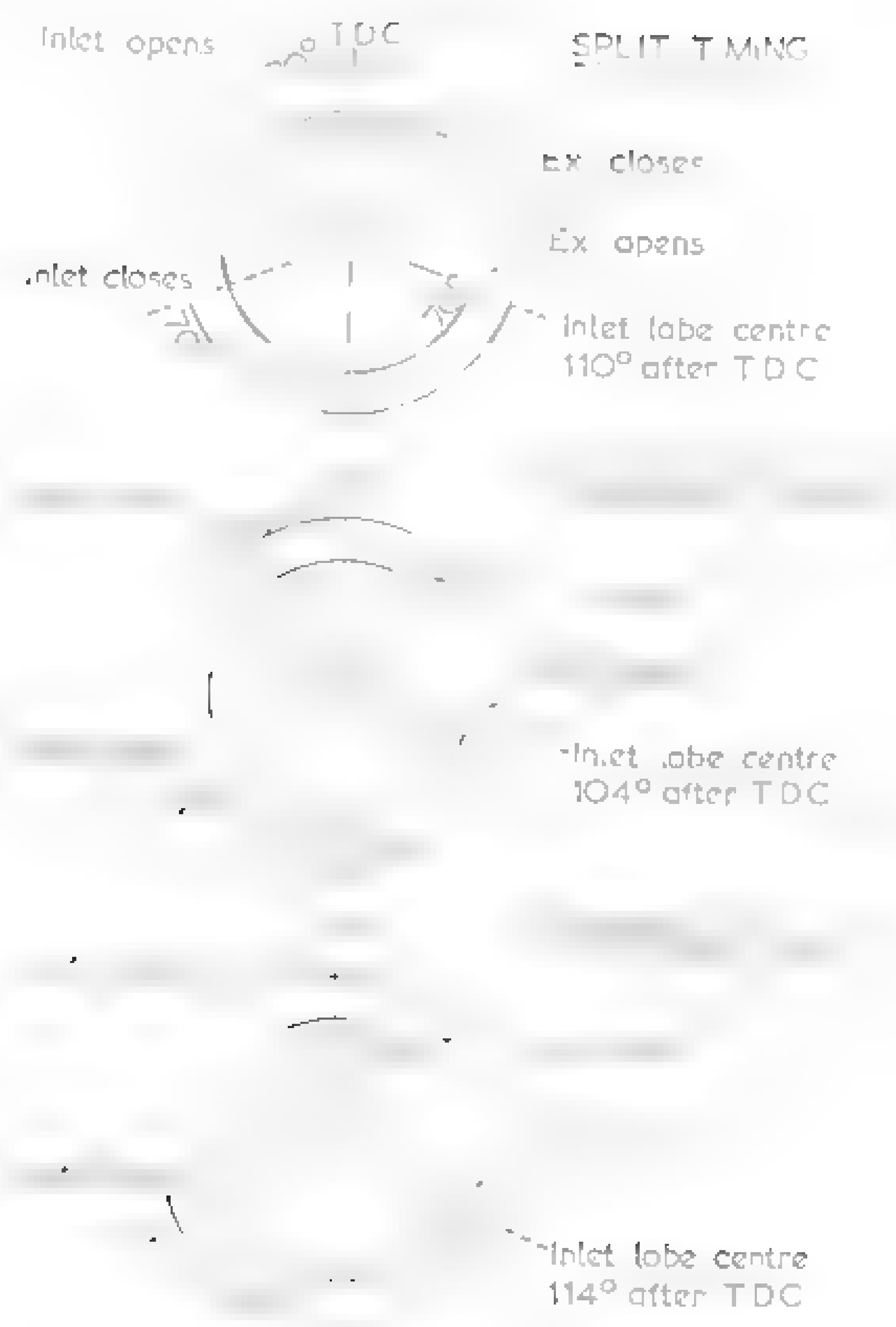


Figure 16.13 Camshaft timing diagrams

You will note that in each instance the duration has remained 280°, the overlap 60°, and the lobe centres 220° apart. This being the case you are probably wondering why bother 'timing' the cam timing if duration, overlap, lift and lobe centres stay the same? If you think about it, of course these figures must stay the same – we are not altering the physical shape of the cam. All we are doing is changing its position in relation to the crankshaft. There is an exception to this, and that is in the case of twin overhead cams, where we may advance the inlet cam, say, 5°, and advance the exhaust cam 9°, but more on that later.

Basically, advancing the cam will improve bottom end and mid-range power,

while retarding it will decrease bottom end and mid range power but may very slightly increase the top end. Around 2° advance will have very little effect, but from 4° to 6° it will show a marked improvement in mid range power, perhaps with some decrease in maximum power. The improvement in the mid-range comes about as a result of a decrease in the reverse pumping action, which raises cylinder pressure. Because the inlet valve is being closed earlier, there is less mixture being pushed back up the inlet port as the piston moves up. As the exhaust valve is also being closed earlier, there is [illegible] as the piston descends. Generally, I prefer to run all road and rally engines with the cam 4° or 5° advanced. For track work, it depends on the nature of the engine. If it is an unrestricted engine I usually run 3–4° advance, but for small-carb restricted, or 9 1 engines 6–8° advance is typical.

I never advise cutting back on cam advance except in one circumstance, and that is when the machine is losing traction because of excessive torque in the low or mid-range. This will usually apply only to drag strip or speedway vehicles. On the strip, too much low-end may cause excessive wheelspin or rear end location problems, while at the speedway an excess of power at a particular engine speed may be causing traction problems on the exit from a specific turn. Taking off 4 or 5° advance should help with either annoyance.

VALVE FOULING IN TWIN-CAM ENGINES

With twin-cam engines we have much more scope when it comes to cam tuning. Not only can we advance or retard both cams by the same angle relative to crankshaft rotation, but we can also alter the lobe separation angle between the inlet and exhaust cams without having to go back to our cam grinder and pay money for another cam. We can choose one lobe profile for the inlet cam and a different lobe for the exhaust cam. In fact, we may decide on an exhaust cam from one cam grinder and an inlet from another. However, caution is necessary to avoid valve-to-valve fouling during the overlap period, especially with the more radical race grinds.

When you start juggling twin-cam timing, keep in mind the need to maintain a minimum of 0.060in clearance between the inlet and exhaust valves. I check this by drawing up the cam displacement curves for both the inlet and exhaust cams. But remember when you draw up the curves that the angles (see Table 10.5) for each 0.010in lift will all have to be multiplied by two, as the camshaft is turning at half the speed of the crankshaft. Therefore for each revolution (360°) the camshaft turns, the crankshaft turns twice (720°). As we set the cam up in relation to the crankshaft, we now have to know what its valve lift is relative to degrees of crankshaft rotation.

To save time I draw both curves on separate pieces of transparent drafting film, to the same scale. This way I can lay the curves over each other according to whatever lobe centre position I wish to check. To keep things accurate, a third piece of paper is required as a reference or standard, and this piece is placed under the two displacement curves. The reference paper will have just three vertical lines on it. The centre will indicate TDC, the first line will indicate the exhaust lobe centre (full lift position), and the third line will show the inlet lobe centre. These lines will be to the same horizontal scale as that of the displacement curve. The lobe centre (full lift)

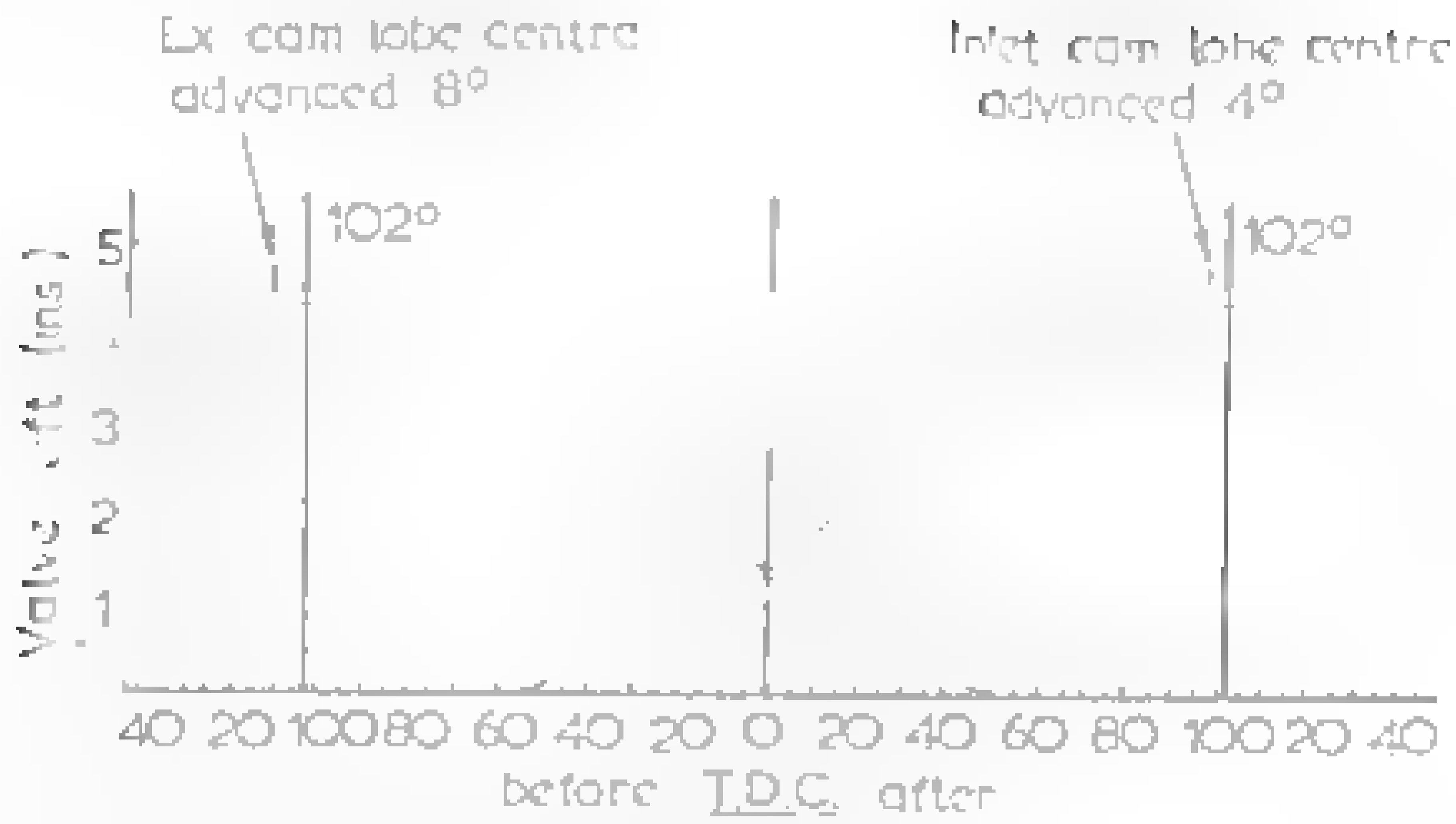


Figure 10.14 Valve interference checking curves.

positions will be marked in accordance with the cam timing figures given by the cam grinder (Figure 10.14).

If the cam grinder has given a lobe centre figure of 102°, this means that the cams should be set up such that the inlet valve is at full lift 102° after TDC and the exhaust valve is at full lift 102° before TDC. However, if no lobe centre angle is stated, you will have to work it out from the timing figures. Let us say that the timing is 53°–77°/82°–58° for the inlet and exhaust. This would mean that the cam had an inlet duration of 53° + 180° + 77° = 310°, and an exhaust duration of 82° + 58° + 180° = 320°. Therefore the lobe centres of the inlet and exhaust have to be 155° (310° ÷ 2) and 160° (320° ÷ 2) respectively after the opening point of each valve. If we subtract the inlet opening angle of 53° from 155°, we find that the full lift (or lobe centre) position will be 102° after TDC. For the exhaust we subtract the closing angle, 58°, from 160°, and find the exhaust lobe centre to be 102° before TDC.

Now that you know the lobe centre angles, draw these in as reference lines before and after the TDC line. If you decide to advance the exhaust cam 8°, lay the exhaust displacement curve on the reference sheet with the peak of the curve (the lobe centre) 8° before the exhaust reference line. Similarly, if the inlet is being advanced 4°, lay the inlet curve with the peak 4° ahead of the inlet reference line. It is only by moving the displacement curves around in relation to the reference lines that the next part of the operation can be accurate.

Looking at the curves it is obvious that the valves will come close to fouling during the overlap period, so what we have to do is make up a list to show us what the inlet and exhaust valve lift will be for each 2.5°. It is not necessary to record the lift over the entire overlap period, usually 30° on either side of the intersection point of the lift curves is enough. Once the list is made up, it is necessary physically to measure the clearance between the valves. If at, say, 10° after TDC the inlet valve has a lift of 0.180in, and the exhaust a lift of 0.156in at the same angle, accurately set the valves up in the head at those measurements and measure the clearance. As you can see, this involves quite a deal of work, but if you are after that last bit of power and you do not want a wrecked motor, it is a must. If it is an engine type you do a lot of work with you can skip a lot of this hassle by sacrificing an engine block and slicing off the rear

cylinder. Then with the head fitted you can fairly easily get into the combustion chamber to physically measure valve head clearances with the cams timed to a variety of lobe centre angles. If you have a lot of faith in computer software you can also simulate this sort of thing, but I sleep a lot easier when I can actually measure things.

The procedure just outlined to check twin cam engines for valve fouling can also be used if you are running a push rod canted valve motor and you wish to increase the rocker arm ratio. Normally, the cam grinder should be able to tell you if you can use that particular cam with, say, 1.65:1 ratio arms instead of the standard 1.5:1 arms, without risk of the valves coming too close together. However, if he is not able to advise you on this, it will be necessary to work it all out as for a twin cam.

HOW TO TIME THE CAM

Now that you know why we advance or retard the cam, you are in a good position to set up your cam just where you want it. With the cam fitted, and the timing marks lined up, the next step is to see where the cam is in relation to the crankshaft. In other words, is it advanced or retarded? To check this you will need a degree wheel (8in is a good size) or a 360° protractor bolted to the nose of the crank. You will also need a dial gauge to measure the cam lift. You should fix a good solid pointer under a convenient bolt to lap over the face of the degree wheel.

The first thing we must do is accurately to find TDC, and for this we need another piece of equipment, a positive stop. If you are finding TDC with the heads [illegible] a piece of steel rod welded in so that it projects past the thread far enough to stop the piston reaching TDC.

An easier way to find TDC is when the head is removed. Then we merely need a [illegible] bolt in the middle of the positive stop bar.

With the positive stop fitted, rotate the crank as far as it will go, taking care not [illegible] two stop angles is TDC, so remove the positive stop and rotate the crank to TDC — the halfway point. Now very carefully loosen the degree wheel and turn it around to align the zero mark with the pointer, and lock it in place. To double-check, refit the positive stop and rotate the crank in both directions as before. When the degree wheel is in the correct position, the number of degrees before and after TDC will be identical when the crank stops.

After TDC is found, you can go ahead and check the cam phasing. It is a waste of time trying to set up a cam according to the cam grinder's timing figures. The best way is work out the full lift (or lobe centre) angle and see at what angle you are reaching full lift. If, for instance, the cam had a full lift angle of 110° after TDC, but maximum lift was occurring at 104° after TDC, it would mean that the cam in that position is advanced 6°.

The cam grinder may supply you with checking (or degreeing) timing figures for 0.050in lift. The cam timing may be 60°–85°/96°–52° and the timing at 0.050in lift 41°–6[illegible]°/77°–33°. Set up the dial gauge on the lifter or push rod [illegible]

record the angles for 0.050in lift. Your cam may check out 41°-70°, 76°-34°, meaning it is retarded 1°.

Having determined where the cam is, you may decide to change its phasing, to tune it to your engine. The next problem is just how to do it. With some motors this is easy as they have vernier timing wheels (eg Jaguar and Alfa Romeo) or taper fit wheels (Ford BDA). The majority, however, use either dowels or keys to locate the cam or crank sprocket positively.

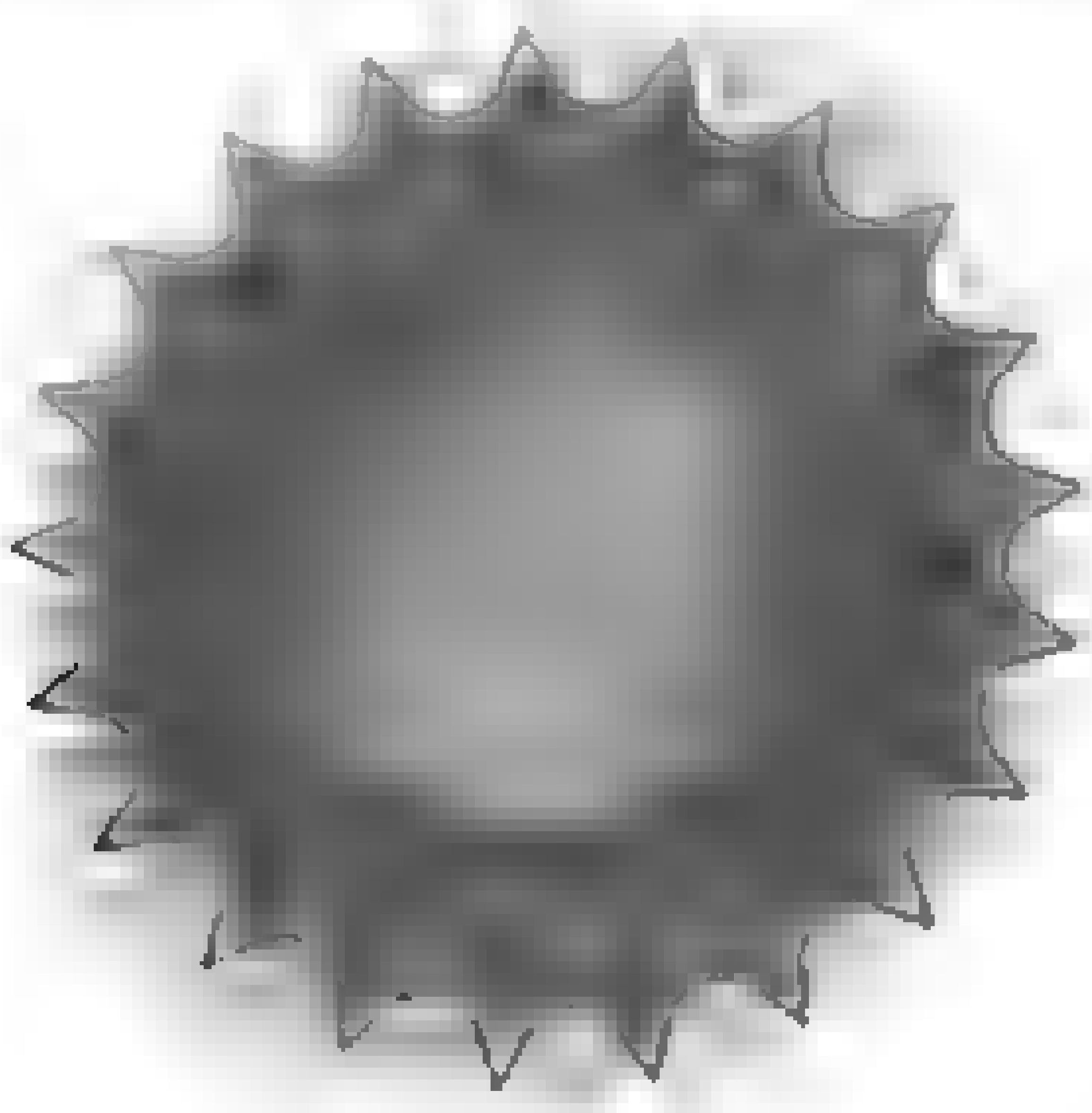
Dowelled drives will need offset dowels fitted and, if more than a central bolt fixes the sprocket to the cam, the bolt holes will require elongation with a round file. The amount of offset required is minute. For example, with the push rod British Fords and Lotus Twin Cam, 0.006in offset (at the cam) represents 1° at the crank, so to change the cam position by 6° (at the crank) requires a dowel with 0.036in offset. Instead of supplying offset dowels, some cam manufacturers have a range of offset bushes that are fitted to the cam sprocket. Others are able to supply slotted cam sprockets, and many V8s have a range of crank drive sprockets available cut with three or more keyways to change the cam phasing.

If there is nothing available for your particular engine, you will have to make up the appropriate hardware. Keep in mind that the dowel or key does not require hardening in most instances as it does not actually transmit the driving force, which is taken by the friction between the sprocket and shaft (key located) or by pressure between the sprocket and cam created by the fixing bolts (dowel-located). Incidently these fixing bolts should be locked using Loctite.

It can be difficult to figure out just which way to move a cam and by how much to advance or retard it. To advance the cam in relation to the gear or sprocket attached to it we have to move the camshaft further forward in the same direction of rotation as illustrated in Figure 10.15. Conversely to retard the cam we move it backward in the opposite direction to the normal rotation of the cam timing gear.

Because the cam turns at half engine speed, it only moves through half the number of degrees relative to the crankshaft. Hence if the cam timing is retarded 6°

This crank timing sprocket has multiple keyways to simplify cam timing changes



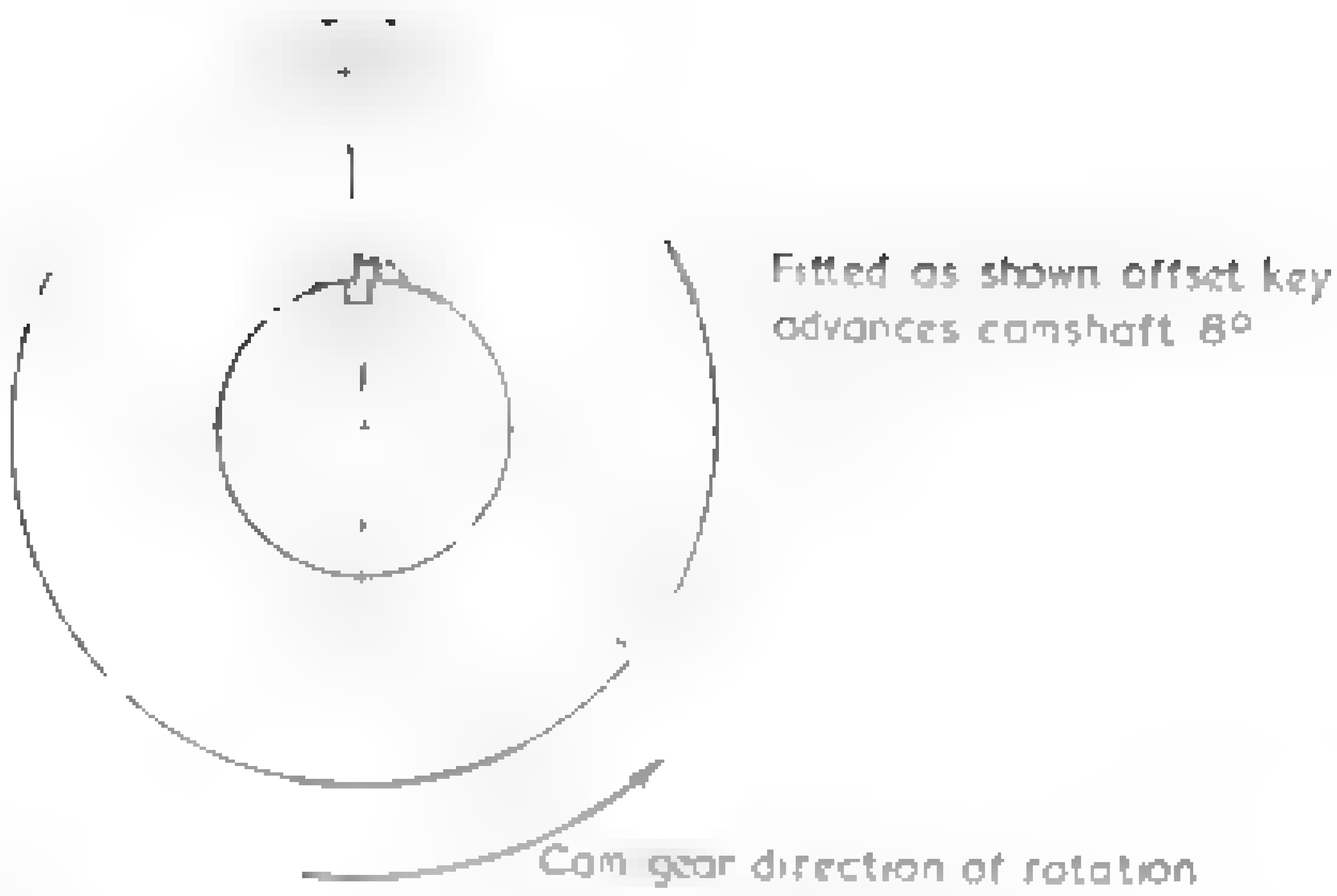
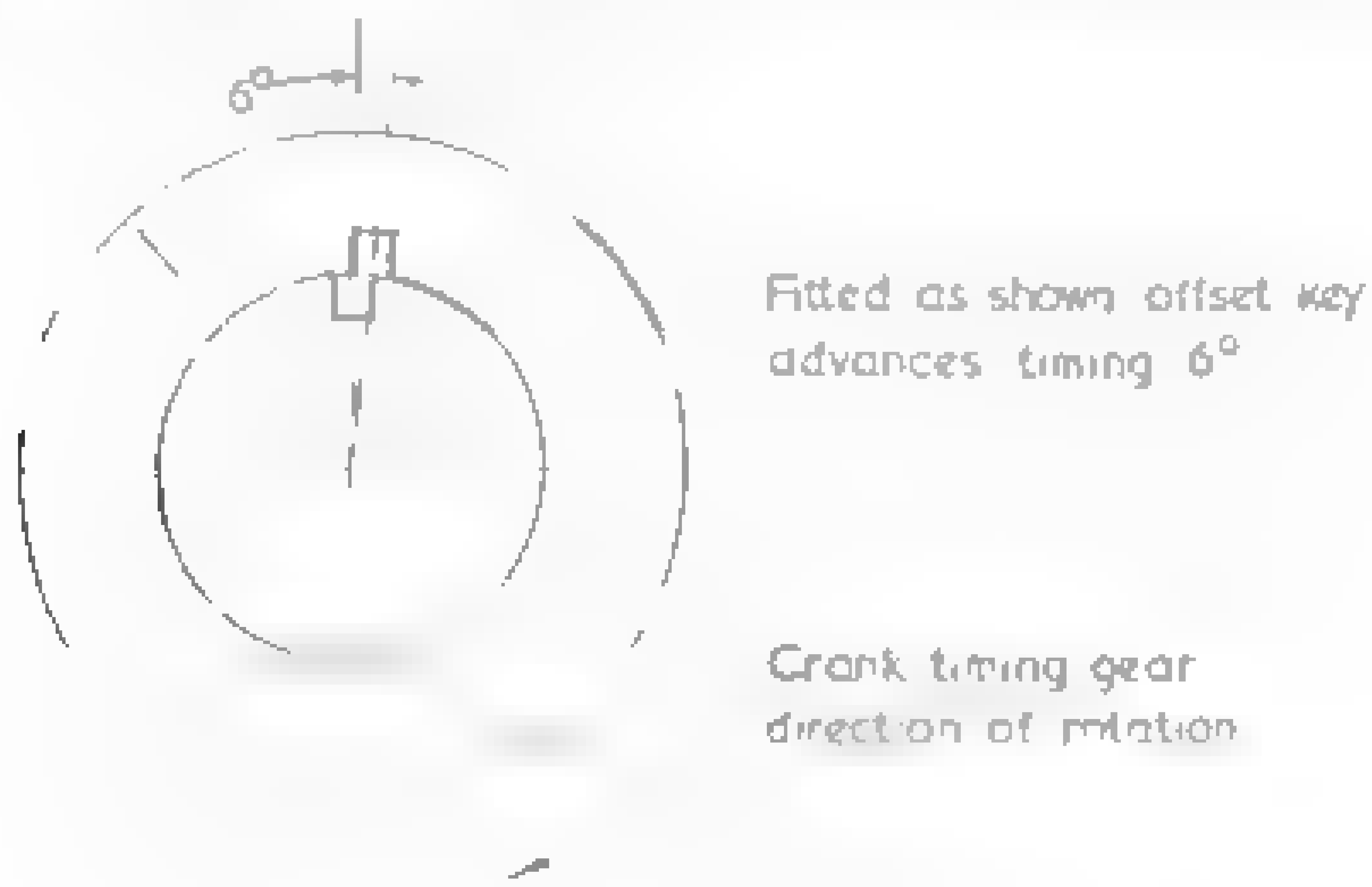


Figure 10.15 To advance the cam it is moved forward relative to the cam gear's direction of rotation

from what we want, then the cam must be advanced 6° measured at the crank, or only half that, 3°, at the camshaft. Therefore if we plan on making the adjustment at the cam and cam timing gear a 3° offset key/dowel will be used.

To calculate just how much the key or dowel must be offset to move the cam 3°, we need to accurately measure the radius between the cam centre and the centre of the [illegible] of 360° the key will move around in a circle with a radius of 0.594in and a [illegible] 3° within the timing gear the key must be offset 0.031in (3.732 x 3 ÷ 360 = 0.031).

Figure 10.16 When the valve timing is advanced at the crank timing gear, the gear has to be moved forward on the crank nose in the direction of normal engine rotation



which will advance the cam 6° when measured on a degree wheel bolted to the crankshaft

If the timing change is to be made by moving the gear fitted to the crankshaft the procedure is different. To advance the cam the crank timing sprocket has to be moved forward on the crank nose relative to the normal direction of engine rotation (Figure 10 16). To retard the timing move it backward The amount the key has to be offset is calculated in exactly the same way as previously, but the offset has to be the full number of degrees, in this example 6°

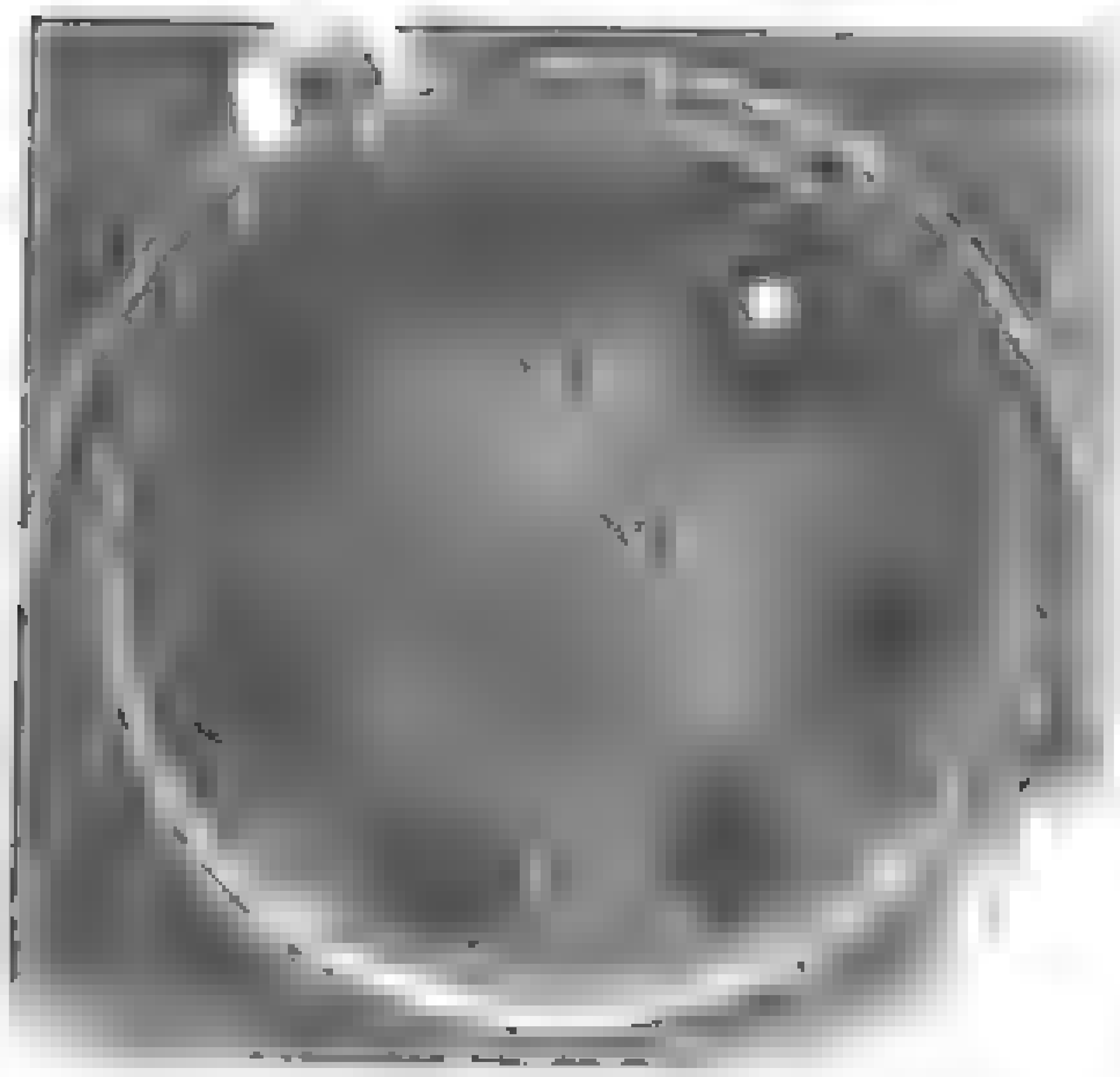
With ohc and twin-cam engines I prefer not to use slotted cam pulleys. I find it a hassle setting up a dial gauge and degree wheel when all I want to do is make a simple cam adjustment to see how the engine responds Some tuners mark the inner and outer ring of slotted pulleys, but unless the cam is being moved something like 6–10° (measured at the crankshaft) this is very inaccurate and not good for repeatability during test sessions. To get around this problem I use multi-hole cam pulleys If I have a pulley with eight holes each 'plus 1°' apart, once I have initially set the lobe centres using a dial gauge I know that moving the cam forward one hole will advance the cam 2°; likewise moving the cam back one hole will retard it 2°. So when the engine is assembled the cam pulley is positioned to give, say, 4° advance using the dial gauge and degree wheel Later, during a dyno session, moving the cam forward one hole in the pulley gives exactly 6° advance Conversely swinging the cam back one hole will retard the cam 2° However, as it was initially set up 4° advanced, it is in reality 2° advanced

It is quite simple to calculate where the dowel holes or keyways should be machined If, for example, the pulley has 40 teeth, this means that each tooth, from centre to centre, is 9° apart (360 ÷ 40 = 9) The first dowel hole, the index hole, should be centred directly under the centre of a tooth The next hole centre will be 5 teeth + 1°, or 46° clockwise from the index hole ((9 x 5) + 1 = 46) The third will be at 10 teeth + 2°, or 92°, the fourth at 15 teeth + 3°, or 138°, followed by the fifth, sixth, seventh and eighth at 184°, 230°, 276° and 322° respectively. Obviously a machine shop carrying out this work must be able to guarantee the accuracy of dowel hole or keyway placement

ADJUSTING VALVE CLEARANCE

With the cam phasing sorted out, the next thing we have to think about is the valve clearance ('lash'). The cam grinder will supply a clearance for the intake and exhaust valve, and sometimes he will specify a different clearance for different types of valve material. If he gives a 'cold' clearance he means just that; the motor should not have been run in the previous 5 hours The 'hot' clearance is for a motor at normal operating temperature, usually 70–88°C water temperature, 90–100° oil temperature It is very important to maintain sufficient valve clearance to allow the valve to seat firmly and to remain on the seat long enough to transfer heat to the cooling medium Running the valves tight to quieten them is a sure way to get valve burning in a performance engine

Sometimes we can benefit by changing the valve clearance within reason and according to the type of competition involved Note that we are talking about a competition motor here, not a road or rally motor. We have already pointed out that for serious competition the motor must be set up to suit the track or stage, but our best [illegible]



A cam drive sprocket with multiple dowel holes allows the cam to be advanced or retarded accurately

plans can go astray. Suppose we have set our cam for a particular dirt speedway but as the night progresses the surface begins to break up or the weather turns foul. Obviously we are not going to have time to advance the cam, so the only way to pick up some bottom-end power in a hurry is to increase the valve lash. This will have the effect of shortening the duration by 3–4° for each 0.005in that the clearance is increased. There is a very definite limit to how far we can go, however, as this has the effect of shortening the opening and closing ramps, which could lead to serious shock loads in the valve train. Therefore the clearance should not be increased by more than 0.005in unless the cam grinder says otherwise. Keep in mind that the valve float limit will now be lowered, so be careful.

If, on the other hand, you find that you need more top-end power on the night, you can decrease the lash, but this is only feasible for short-distance events such as drags, hillclimbs or speedway sprints. Remember that decreasing the lash will increase valve temperatures, so always maintain 75% of the recommended clearance. Also, if you know that the valves and seats are getting a bit tough or the guides are shot, do not decrease the clearance at all – you could easily wreck the motor. An increase in valve temperatures, under competition conditions, could cause pre-ignition or detonation: both are piston and engine breakers.

With push rod type engines you must maintain a valve-to-piston clearance of 0.080in and 0.100in for the inlet and exhaust valves respectively. Therefore it will be necessary to set up the pistons with adequate valve cut-outs before you attempt to change the valve clearance or change the cam phasing.

It is surprising how many enthusiasts have little idea of setting valve clearances accurately. First, let me point out that it is impossible to do much of a job if the contact face of the rocker or valve stem tip has a dip worn into it. The feeler blade will be too wide to fit into the dip, so we will not be able to take this wear into account. When we measure the clearance we want to have the lifter fairly close to the centre of the bas

circle. With most ohc engines it is possible to see when the lifter is on the base circle but with other types we have to use another method.

The simplest method for in-line engines is visually to divide the engine in half. All the rockers in the front half we number from 1 to 4 (or 1 to 6 if it is a six-cylinder) and we do likewise for the rear half, but this time we start No 1 from the extreme rear of the engine. In this way we have both No 1 rockers at opposite ends of the head, and both No 4 (or No 6) rockers in the middle, alongside each other. As we turn the motor over we always adjust the valve that partners the one that is fully open. Therefore if No 1 at the rear of the motor is fully open, we adjust its partner No 1 at the front of the motor; if No 3 at the front is fully open, we adjust No 3 at the rear, and so on.

With flat and vee motors, the safest way is to turn the motor over and adjust the inlet and exhaust rockers for each cylinder together, midway between the inlet closing point and the exhaust opening point, ie with the piston at TDC at the end of the compression stroke. If you adjust each cylinder in turn according to the firing order, it will save turning the motor over unnecessarily.

When adjusting clearances on ohc motors, I have found it simpler and more accurate to adjust the clearance by inserting the feeler strip between the cam and rocker rather than at the valve stem. However, when using this method it is necessary to reduce the clearance because of the effect of the rocker arm ratio. Therefore if the clearance at the valve is normally 0.013in and the rocker arm ratio is 1.45:1, the clearance at the cam lobe will be 0.013 ÷ 1.45 = 0.009in.

ADJUSTING VALVE CLEARANCE WITH HYDRAULIC TAPPETS

The majority of manufacturers recommend that a hydraulic valve train mechanism is adjusted hot, with the engine running. They suggest that each rocker is slackened off until it emits a distinctive 'clacking' sound, then the adjuster should be taken down one half turn. What I prefer is to carry out the adjustment with the engine stopped and either cold or warm. With the lifter on the cam base circle, slacken off the rocker to give a little clearance at the valve. Then rotate the push rod and tighten the rocker down at the same time. When you feel some resistance to turning the push rod, note the position of your spanner and tighten the adjusting nut a further quarter-turn.

Today, many manufacturers make no provision for the adjustment of hydraulic valve systems, so when you fit a cam with a smaller than standard base circle you will have to take special steps to obtain the correct adjustment. The simplest but most expensive way out is to fit adjustable push rods. A fairly effective and less expensive method is to machine the base of the rocker pedestals by an amount equal to half the difference of the base circle diameter of the standard and modified cam divided by the rocker ratio. For example, if the standard base circle diameter is 1.260in and modified it is 1.120in, and the rocker ratio is 1.5:1, the pedestals will require 0.047in milled off their bases (1.260 − 1.120 = 0.140, 0.140 ÷ 2 = 0.070, 0.070 ÷ 1.5 = 0.0466). Often, though, the head or block will have to be machined to increase the compression ratio, so this has to be taken into consideration. If in the above example 0.060in had been taken off the head, the pedestals would have to be shimmed up. The most accurate method to determine the required shim thickness is to tighten down each pedestal a quarter-turn past the position where resistance to turning the push rod is felt. Now measure the clearance under each pedestal and insert a shim of the same thickness.

Chapter 11

The Bottom End

Enthusiasts associate high performance with cams, carburettors and exhaust modifications. However, many hidden horsepower can be found by proper attention to blueprinting the bottom end of a motor. Remember that it is in the bottom end where we lose the most power to friction. Consequently everything we do to improve the accuracy and fit of components gives us back power that would otherwise have been burned up fighting friction. This is also the place where if we have the budget we can further minimise such losses by the application of the new high tech coatings. An additional consideration is the added reliability we can expect when things are done properly and to an exacting specification.

Logically, the place to start is the cylinder block, because it is the home for everything else in or on the engine. First, inspect the water passages for rust or scale. If any is present it will be necessary to have the block boiled and cleaned in a chemical bath. This should be done with all the welch plugs (freeze or core plugs) and oil gallery plugs removed.

A mildly dirty block can be steam cleaned or washed with solvent. At the same time the oilways should be carefully cleaned, using brushes designed for this task. Stud holes deserve the same treatment, particularly head and main bearing cap stud holes. Finally, high-pressure air must be used to clean and dry off everything. Even brand new (green) blocks should be cleaned as outlined, with particular attention to ensure that no casting sand has been left in the water passages.

INITIAL BLOCK PREPARATION AND CRACK TESTING

With the block perfectly dry, it can be visually inspected for cracks. A competition block should be pressure crack tested, really it is a waste of time and money if this is not done first. You can do a pressure test yourself as only very basic equipment is required. Preferably the head should be bolted to the block (it is acceptable to use an old head gasket for this test if it is not torn) and both head and block should be fully stripped, but with welch plugs in place. All water openings should be blocked off with steel plates

One plate should have an air line fitting installed (or you may prefer a tubeless tyre valve). Connect to this your air line tap and pressure gauge (the use of an old oil pressure gauge is suitable). Now slowly open the air line valve and gradually let the air pressure come up to 40psi. Do not let pressure come up too quickly because if a welch plug is not in tight or if there is a weak spot in the block it could blow out and cause you injury.

If everything is holding satisfactorily, again open the air valve and let the pressure slowly build to 50psi. Some people advocate that you should check at 60psi, but this can be quite dangerous if there is a weak spot, and in any event most cracks will become evident at 45–50psi. (Note that heavy duty thick-wall race blocks cast with special chrome moly alloy iron can be tested up to 100psi.) It is really more a matter of how carefully you look for cracks as you slowly work your way over the block (check the head too, and inside the inlet and exhaust ports), rather than going to very high air pressure.

To make the cracks more clearly visible some prefer to spray over the entire [illegible] covering just a small area at a time, while others choose to spray on water with a little added liquid detergent. Either way any cracks will be shown up by the presence of air bubbles.

Note that if you find a crack do not quickly discard the block unless it is obvious that it is not repairable either because the crack is too large or it is in a critical location. Some cracks can be inexpensively repaired, and a block that has been repaired may be just as strong as one that is not cracked. In fact, some engines routinely crack in certain areas, so if you are too fussy you may have to check out many second-hand blocks to find one that is not cracked, then after you race it a few times you may find that it too has cracked, whereas the block that was initially inspected and subsequently repaired may now be stronger in that particular spot and so not give any more trouble.

With the block crack-tested, next take a plug tap and clean the head stud and [illegible] reading. Any stud holes that have not been chamfered should be, to prevent threads [illegible] pulled main bearing cap thread can allow a bearing to turn. Finally, check the depth of each hole to be certain that the stud will not bottom.

To avoid cuts to yourself and possible engine damage, carefully grind away any casting slag. The main area for concern is around the main bearing webs, the sump pan deck and, in vee motors, around the oil drain back holes and the rest of the valley area.

MAIN BEARING ALIGNMENT

The next operations involve 'squaring' the block. A check must be made to ensure that the main bearing bores are perfectly aligned, as any misalignment will wreck the bearings and possibly even the crank. It will also soak up a lot of power as the frictional losses are greater. I have found new blocks with up to 0.007in misalignment, so do not assume that the alignment is correct. Any misalignment can be corrected by line-boring. When main bearing caps are replaced by heavy-duty items, line boring will also be necessary. To ensure that alignment is maintained, the main bearing caps should be numbered and have the front position marked. This will assist in fitting each cap in its correct location each time the motor is rebuilt.

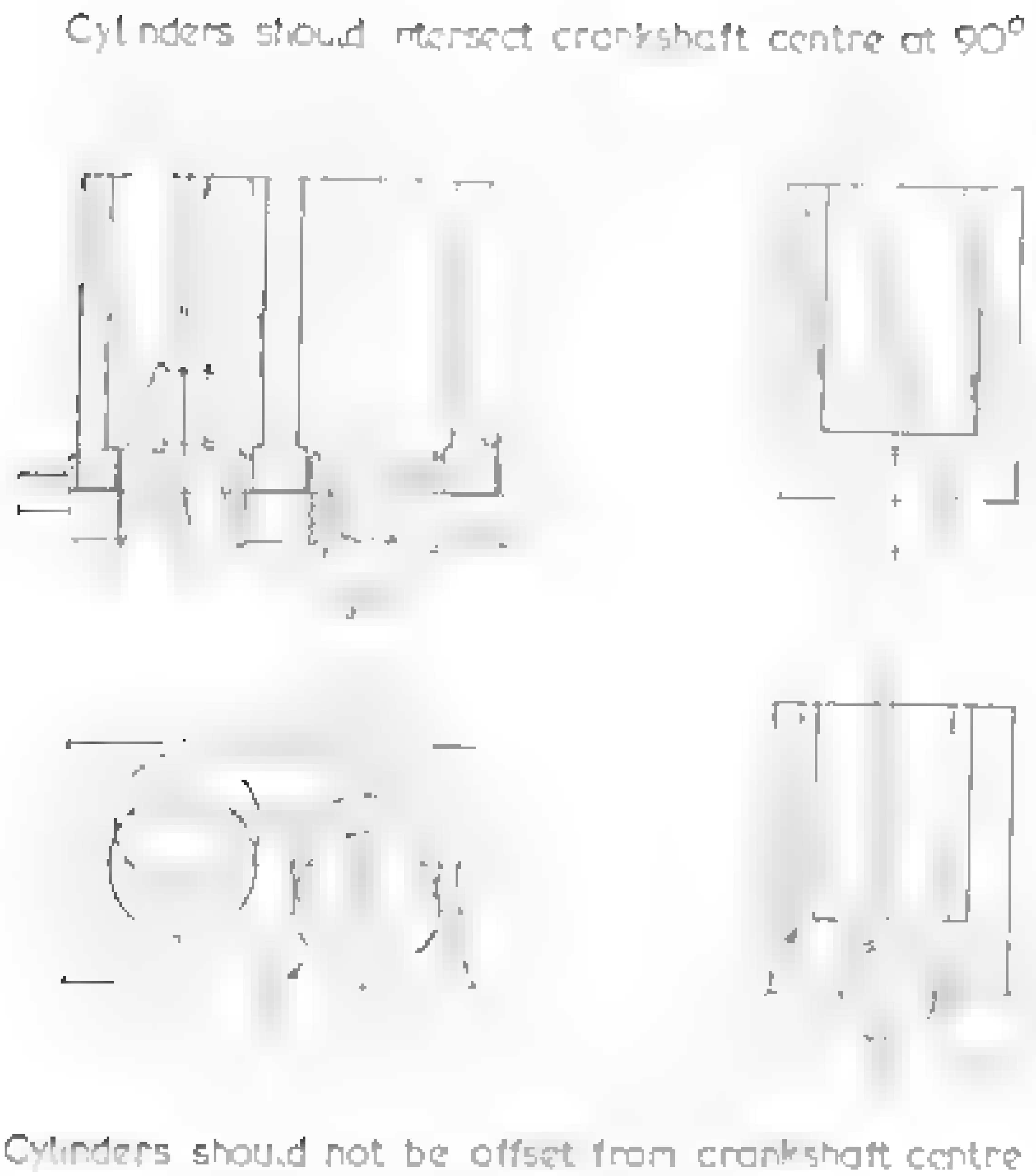


Figure 11.1 Cylinders must be accurately rebored

With the main bearing bores true, the block should be checked at each corner to guarantee that the distances between the top of the block (the deck) and the crankshaft centreline are identical. If the block is out of true, milling will be required. Even if the

the piston and the top of the block, to reduce the squish clearance.

CYLINDER BORING

The cylinder bores must be true at 90° to the crank centreline (ie across the motor) to keep the frictional losses low and maintain good ring sealing. If the bore is canted slightly to the front or the rear, the piston pin will hammer out, so a check must also be made in line (ie at 180°) with the crank. There must not be any steps or taper from the top to the bottom of the bore (Figure 11.1).

Boring and honing should be carried out by a firm with precision equipment able to maintain a tolerance of 0.0003in. Forget about garages using small boring machines that bolt on to the block, as the tolerance can be as bad as 0.002in.

On a street engine the machinist sets up the boring bar centred on the existing cylinder bore. However, there is no guarantee that the existing bore centres are correctly located (indexed) in relation to the crank throws or the combustion chambers.

in the cylinder head. Therefore in a race engine I prefer to see a bore index plate bolted

correctly located over the cylinders.

When a race engine is being bored I like to see only 0.010in taken out with each

CYLINDER WALL THICKNESS AND RING SEAL

stronger and less prone to flexing, due to the use of a special grade of cast iron.

instance.

block rigidity.

power of the engine due to inadequate ring seal.

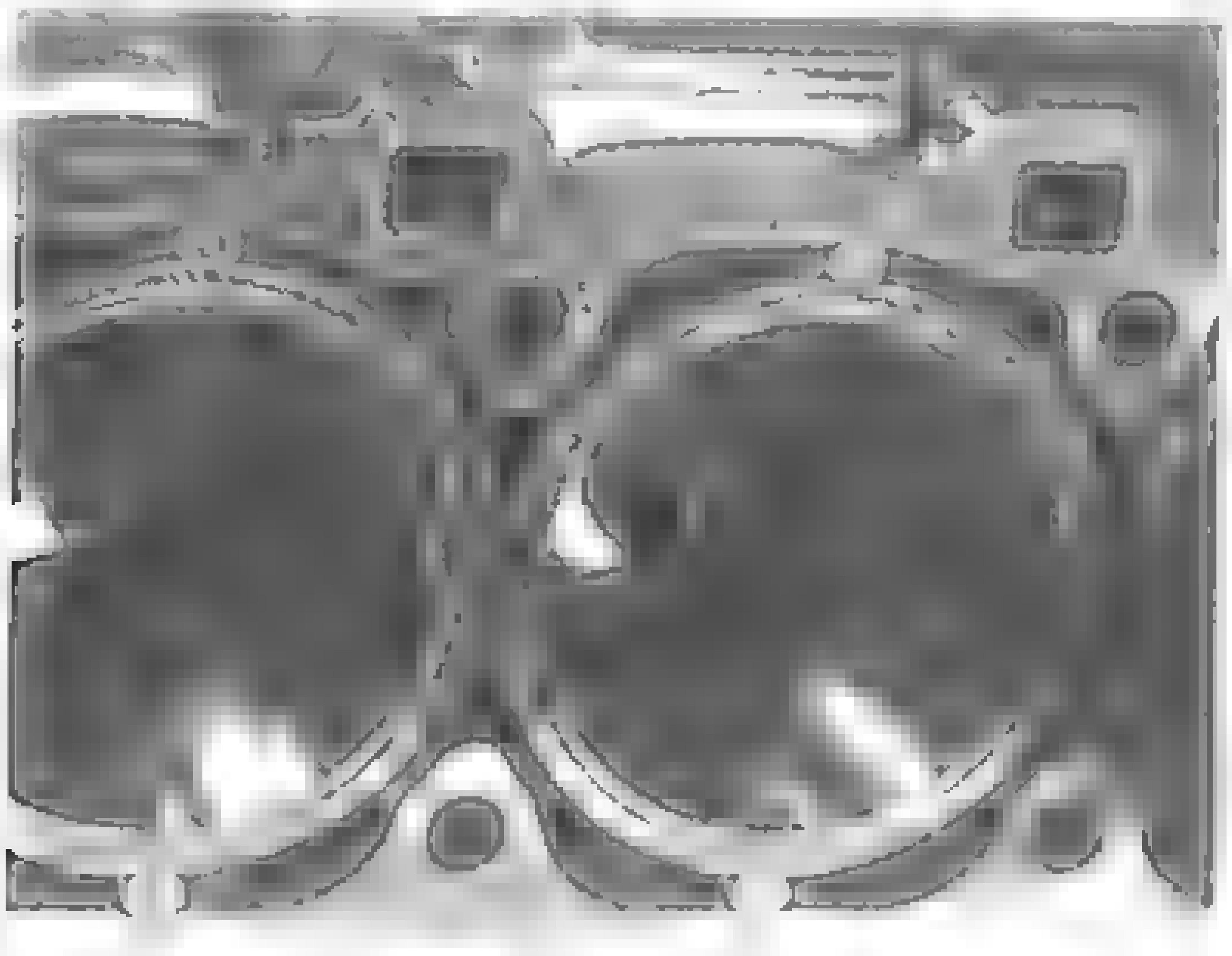
Basically for good ring seal I reckon that you need a cylinder wall thickness of 0.095–0.105in minimum for an engine expected to make 65–70hp per cylinder. This is most critical at the top half of the stroke because it is the top area of the cylinder where we require good ring seal, first to make compression, then, after ignition, to retain cylinder pressure to drive the piston down and make power. Toward the bottom of the cylinders we may be able to drop approximately 0.020in in wall thickness, but there are many other factors to consider such as operating rpm, crank stroke, main bearing web strength and whether the engine is ever likely to suffer detonation.

As the power levels increase, so does the need for thicker cylinder walls. At 75hp per cylinder I would be looking at a minimum wall of around 0.110–0.130in; at 85hp the wall should be 0.150–0.180in and at 100hp 0.220–0.250in. These latter two thicknesses for engines at power levels of 85 and 100hp per cylinder might appear excessive, but if you want a block to seal well and have a service life of up to 5,000 race miles at 8,500rpm, that is the sort of wall thickness you must be considering. I know that some race teams get away with less wall, but I also know that they are scrapping blocks with less than 300 miles on them. Considering the amount of time and money invested in a race block, this does not make much sense, and I wonder how often they lose an expensive crank because of the main bearing webs breaking away.

STRENGTHENING OPEN-DECK BLOCKS

A few engines though have problems that go well beyond cylinder wall thickness. Some for example have an open-deck design where the top of the cylinder is unsupported or only lightly braced by the top of the block. At low power levels this may not cause problems, but as the hp and rpm rise so does cylinder and block flexing. The solution for small Hondas is to fit a block guard. This is inserted into the water jacket to fit between the block and cylinder to add rigidity.

Inserting rods between the cylinders and block serves to brace the cylinders, reducing blow-by and head gasket leakage in high hp engines.



At times the fix can be more radical, and expensive. Some Subarus have an open deck block. Upgrading to a closed-deck block means a big outlay. The K-series Rover 1800 engine has a different problem. As we approach 200hp the engine does nasty things like splitting the cylinder liners. The solution here is to shrink a steel sleeve around the outside of the liner up toward the top.

CHECKING CYLINDER WALL THICKNESS

Checking a block for wall thickness and core shift takes a lot of time and care. However, the reward is that the engine will produce more power and you will not be wasting money on machining a 'thin' block that you will have to scrap after just a few races. I have seen some people trying to figure out cylinder wall thickness in the strangest of ways in the hope of getting a massive overbore. Some apparently feel that they can simply measure the thickness of the web between adjacent cylinders. Then with the welch plugs knocked out, they use a bunch of feeler gauge strips to measure the width of the water jacket between those cylinders, then calculate the wall thickness. Unfortunately, they forget that core shift makes such measurements pretty well meaningless.

Some race teams rely on sonic testing to provide reasonably accurate information on metal thickness. The main thing with sonic testing is for the operator to use care and good judgement, but it is also wise to recognise that a sonic test will not detect a

Measuring cylinder wall thickness has become easier in recent times with the availability of relatively inexpensive sonic testers.

porous area in a block or an area of 'dirty' metal. Also, the actual metal thickness may be 0.020in less than the tester is indicating.

Therefore along with sonic testing it is vital to do a pressure crack test as discussed earlier, and it is wise to measure the block using basic measuring tools. In fact, this is what race engine builders had to do before cheap sonic testers became available. I do not know of any machinist's callipers that will do the job, so you will probably have to build your own. What you need is a pair of vernier callipers with a set of very long jaws. One jaw has to be the correct size and shape, and with just enough 'kick' in the leg to fit down through water jacket holes in the deck of the block. I've seen some people cut up a pair of 6 or 8in outside callipers and weld the legs to the jaws of their verniers, but I generally find that outside callipers have too much 'bow' in the legs to fit down into the water jacket.

What I prefer is to chop up two pairs of callipers, one an inside calliper and one an outside calliper, then bolt one leg of the outside calliper correctly orientated to one jaw of the vernier. Then I reverse one leg of the inside callipers and bolt it to the other jaw of the vernier. With a bit of grinding this leg will fit down into the water jacket through holes in the deck of the block, and it generally has enough kick to contact the outside wall of the cylinders. With these legs bolted, rather than welded, to the vernier callipers it is possible to swap them about or to pull them off for grinding to fit down into the water jackets of different blocks.

BIG CYLINDERS – EFFECT ON POWER

Even when an engine is not excessively overbored there is no guarantee that you will achieve a massive increase in power. I think most people understand that a 10% increase in cylinder displacement will not yield a corresponding rise of 10% in maximum horsepower. However, few tuners realise just how ill-advised large capacity increases can be in some situations. The point to be remembered is this: it is no use giving the engine big lungs if the induction system (ie carburettor, manifold, ports and valves) does not have the capability, or the potential capability, to flow sufficient air to fill those big lungs.

A situation like this is not really a problem if the engine is running in a street car, but in a road race vehicle, and particularly in a rally car, this type of over-modification must be avoided because of the disastrous effect it has of compressing the power band. True, the increase in mid-range power is a very important plus in favour of a displacement increase, but if the engine does not have the breathing ability to be able to rev as hard as previously, the increased number of gear changes now necessary may easily reduce the performance level of the car/engine/driver combination to what it was before the capacity jump.

In Table 11.1 you can see how the power range was affected on an ohc Vauxhall rally motor when the capacity was raised from 2,279cc to 2,496cc, about a 10% increase. The smaller motor makes good power between 4,500 and 7,500rpm, and if need be it can be lugged down to 4,000rpm, which gives a very nice 3,500rpm operating range. The 2.5-litre engine is working very well at 4,000rpm (15hp increase over the 2.3 litre engine), but it runs out of steam at around 6,500rpm, which means that the power band has been narrowed down by 1,000rpm. You will note too that there is only a difference of 1hp in maximum power between both engines.

Table 11.1 Dyno test of Vauxhall ohc rally engine

	2,279cc engine		2,496cc engine	
rpm	hp	torque (lbf ft)	hp	torque (lbf ft)
3,250	100	161.6	103	166.4
3,500	108	162.1	118	177.1
4,000	127	166.7	142	186.4
4,500	136	158.7	163	190.2
5,000	170	178.5	183	192.2
5,500	185	176.6	194	185.2
6,000	199	174.2	206	180.3
6,500	202	163.2	200	161.6
7,000	205	153.8	178	132.6
7,500	197	137.9		

Table 11.2 Dyno test of Vauxhall twin-cam 16-valve rally engine

	2,279cc engine		2,496cc engine	
rpm	hp	torque (lbf ft)	hp	torque (lbf ft)
4,250			139	171.8
4,500	144	156.4	154	179.7
5,000	173	181.7	195	204.8
5,500	188	179.5	212	202.4
6,000	216	189.1	236	206.8
6,500	231	186.6	251	202.8
7,000	240	180.1	244	183.1
7,500	244	170.8	234	164.2
8,000	248	156.2		
8,250	232	147.7		

In its present form the 2.5-litre motor would not be any better than the smaller e in a rally car. However, with a lot of development the big motor proved to be marginally superior. A wilder camshaft raised maximum power to almost 209hp and enabled the motor to run to 7,000rpm (192hp), but the mid-range was so badly affected that the power band was narrowed to even worse than before.

After this a 16-valve Lotus head was tried to determine what effect better breathing would have (Table 11.2). The small engine has a 3,250rpm power range extending from 5,000 to 8,250rpm, while the 2.5-litre motor is best between 4,750 and 7,500rpm, a 2,750rpm spread. When a wilder inlet camshaft was fitted (no other changes) this motor ran much better at 7,500rpm (239hp) and made 230hp at 7,750rpm with just a small decrease in mid-range (190hp at 5,000rpm). Maximum power went up to 253hp.

Obviously in this instance the 2,496cc motor should be superior, but since that time some competition rules have outlawed the Lotus head and the 16-valve Vauxhall head will not perform anywhere near as well (ie 20hp less for the 2.3-litre motor).

Of course not every type of competition machine requires an engine with a good wide spread of power, so large increases in displacement can be made without adversely affecting performance. Therefore if you are involved in rallycross or quarter

...itre Vauxhall in either 8-valve or 16-valve form should ... r. However, as you can see from both tables, there is no significa... ... power, so to ga... ... big motor the suspens... will h...e to be capable of getting those extra 15–20 mid-range horsepower down on the track to accelerate the car faster.

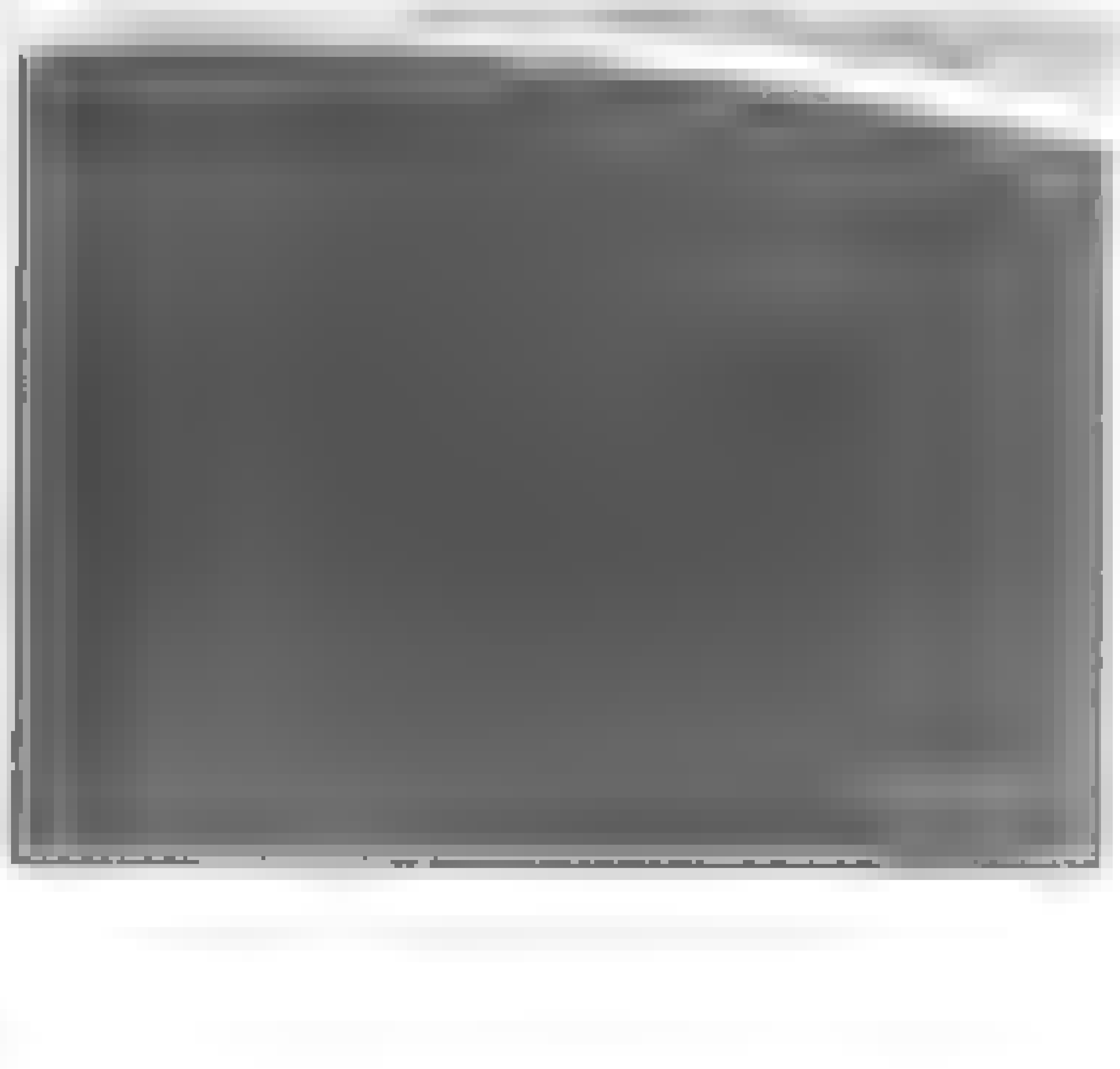
CYLINDER HONING

...ng cylinder honing operations ... te should be fitted with the he... sket, and tensioned to normal head stud tension. This o... necessary on racing engines to guarantee ... es, as the pull of head studs always causes a certain degree of cylinder d... When a honing plate is fitt... the specified head tension, the block ... during the cylinder finish... procedure. Therefore the bores will be tr... head is fitted. The main bearing caps, when tensioned, also distort the cylinders, so these should be fitted as well.

When we say the bores will be true that is in a relative sense only. Meas... the same cylinder bore temperature and block temperature as when honed the bor... will be true. However, once heated to normal race engine coolant temps the bores will no longer remain perfectly concentric. And the ch... will be more extreme in poor quality blocks, excessively overbored blocks, blocks lightened after being hon... on blocks which were incorrectly honed using steel honing plates inst... aluminium (aluminium blocks and cast iron blocks which will be fitted wit... aluminium heads should be honed using aluminium hone plates). The only way we currently have of minimising this inaccuracy is to run hot coolant through the block

...nder trueness and ...

...rrectly honed cylinder an even crosshatch pattern ... waviness or boring tool marks underneath, should ...



during the honing process so that everything is at about 90°C (some prefer higher temperatures; up to 120°C). This isn't perfect either as we can't simulate bore distortion due to combustion pressures and vibration, but it does get us closer to what we are trying to achieve, and this shows in a small power gain of about 1%. Some tuners are finding closer to 2% but my feeling is that since going to hot honing they are now paying a good deal more attention to other aspects of cylinder trueness that were previously neglected, and that is where they are finding the additional hp. Looking at the gain from the aspect of actual blow by you can expect a reduction of around 3cfm.

If you are preparing engines for a category of competition where a 1% power gain is important for a modest additional cost in time and equipment then here is how you need to do it. Apart from the obvious things like a hot water supply and fittings for the block you have to install a very good ventilation system to get rid of all the smoke, and you will do more consistent work using an infrared heat gun to monitor block and cylinder temps. Most cylinders grow 0.0035in when heated from room temp to 90°C so to avoid having the pistons stuck in the bores when the engine is cold you first need to measure the bores cold and then after you have heated the block, before you start honing. Subtract the cold figures from the hot size, and add the result to the dial bore gauge finished size. If you find a cylinder that grows way out of line with the rest you probably have a dud block so don't waste time and money on it. Remember during all this to continually keep checking and adjusting your bore gauge. It will quickly pick up heat and expand. Finally, don't panic if after hot honing you find the bores are all over the shop with the block cold. Of course they will be, but once the block is heated they will again be nice and round.

I prefer to bore the cylinders to within 0.004in of the finished size, then hone each cylinder to fit a particular piston. In this way I can keep piston clearances to a tolerance of about 0.0003in.

The actual honing procedure varies a little from engine to engine, but a fairly standard method using the preferred Sunnen CK-10 automatic hone is as follows. First using 220 grit stones, about 0.003in of material is removed to bring the bore to 0.001in from its final size. Then another 0.0005in is taken out with 280 grit stones, and finally 400 grit stones are used to bring the bore to its final diameter. This results in a very accurate bore with a fine finish that requires very little break-in.

The cross-hatch pattern left by the hone on the cylinder walls is critical. It must be just right if the piston rings are to bed in quickly and have a long life. I prefer a 45° cross-hatch with a finish of 10–12 micro-inches. This type of finish makes it necessary to run the rings in, but they last for a long time and seldom leak. A finish any smoother will not hold enough oil and will allow a glaze to form on the ring face and bore wall. Oil consumption will be a problem and power will be lost due to blow-by. A rougher finish will usually eliminate the need to bed in the rings; however, ring life is greatly reduced. Keep in mind too that glazing can again be a real problem, but not due to lack of lubrication; a rough finish acts like a file on the rings, and the added friction increases their temperature and allows glazing to form.

The upper and lower lip of each cylinder should be chamfered lightly to remove the sharp edge produced by boring. A smooth-cut half round file will do the job, but take care not to let it slip and nick the cylinder wall. While you have the file at hand lightly dress the sharp edges of the main bearing caps and webs.

SPECIALISED CYLINDER COATINGS

With increasing recognition of hp losses due to friction it is obvious that one of the major areas of friction within an engine – between the piston rings and cylinder wall – has to be carefully scrutinised. Originally, coatings were applied to provide a superior bore surface that improved ring seal and reduced bore and ring wear rates. Going back about 30 years Mahle began working with a superior electro-chemical plating called Nikasil. This plating was originally developed for Mercedes when they were building experimental Wankel rotary engines. Then Porsche began using Nikasil plated cylinders in the 630hp air cooled 917 model Le Mans racer. This engine later produced 1,100hp in turbocharged form for the Can-Am series. Today, Nikasil cylinders are in use on tens of thousands of chain saws and other industrial two-strokes throughout Europe. It has proved to be very successful in racing two-stroke engines as well. In time it began to be applied directly to the aluminium cylinders of four strokes, initially as a better alternative to chrome plating of linerless bores, and later because of its ability to also reduce frictional losses.

The Nikasil coating is a nickel and silicon carbide matrix about 0.07mm thick. The nickel matrix is very hard, but it is comparatively ductile, whereas chrome is brittle. Dispersed through the nickel are particles of silicon carbide less than 4 microns in size. These extremely hard particles make up about 4% of the coating and form a multitude of adhesion spots on which oil can collect. So besides providing a very long-wearing surface for the piston and rings to bear against, the silicon carbide particles also contribute to long engine life by ensuring good cylinder lubrication.

Naturally other companies weren't going to sit back and let Mahle steal the market so they began to develop their own proprietary coatings. Consequently there are now many coatings available, even from within the same company, so there are numerous products/processes from which to choose.

A joint project of Perfect Bore Liners and Poeton Aptec is their ceramic coating, Apticote 2000. In regular use now since late 2000, it consists of a hard metal matrix packed with ultra-hard ceramic particles. The hard metal coating has a strong affinity for holding oil which allows for an extremely fine hone finishing of the cylinders, down to as low as 0.05Ra. This reduces ring and bore wear, improves gas seal and oil consumption, and cuts frictional losses. Some are reporting power improvements of as high as 1.5%, but I find a little under 1% to be a more realistic expectation. The reduction though in ring, piston and cylinder wear is remarkable, with bore wear cut to about 20% of what is typical in a race honed cast iron cylinder. Some engine builders say that another advantage of this product is that because of Perfect Bore's expertise with very thin, down to 0.75mm thick, dry aluminium liners – they also have steel and MMC available – engine blocks do not have to be shipped all over the world to be coated/recoated. Blocks can be prepared to Perfect Bore's specs and then have finished liners pressed into place. The disadvantage of doing it this way though is that you are never going to be able to achieve the quality of bore finish and accuracy that Perfect Bore can deliver when they do the work in their temperature controlled environment using the best honing and measuring gear available. While you will no doubt get similar wear improvements with the pressed in liners, I don't think it would be possible to arrive at the same precision seal between piston ring and bore, so I wouldn't expect an hp gain in a fresh engine over that possible with a careful hot hone

of uncoated bores. However, at the full-life stage I would expect it to show ... lower than an engine with uncoated cylinders.

FINAL BLOCK PREPARATION

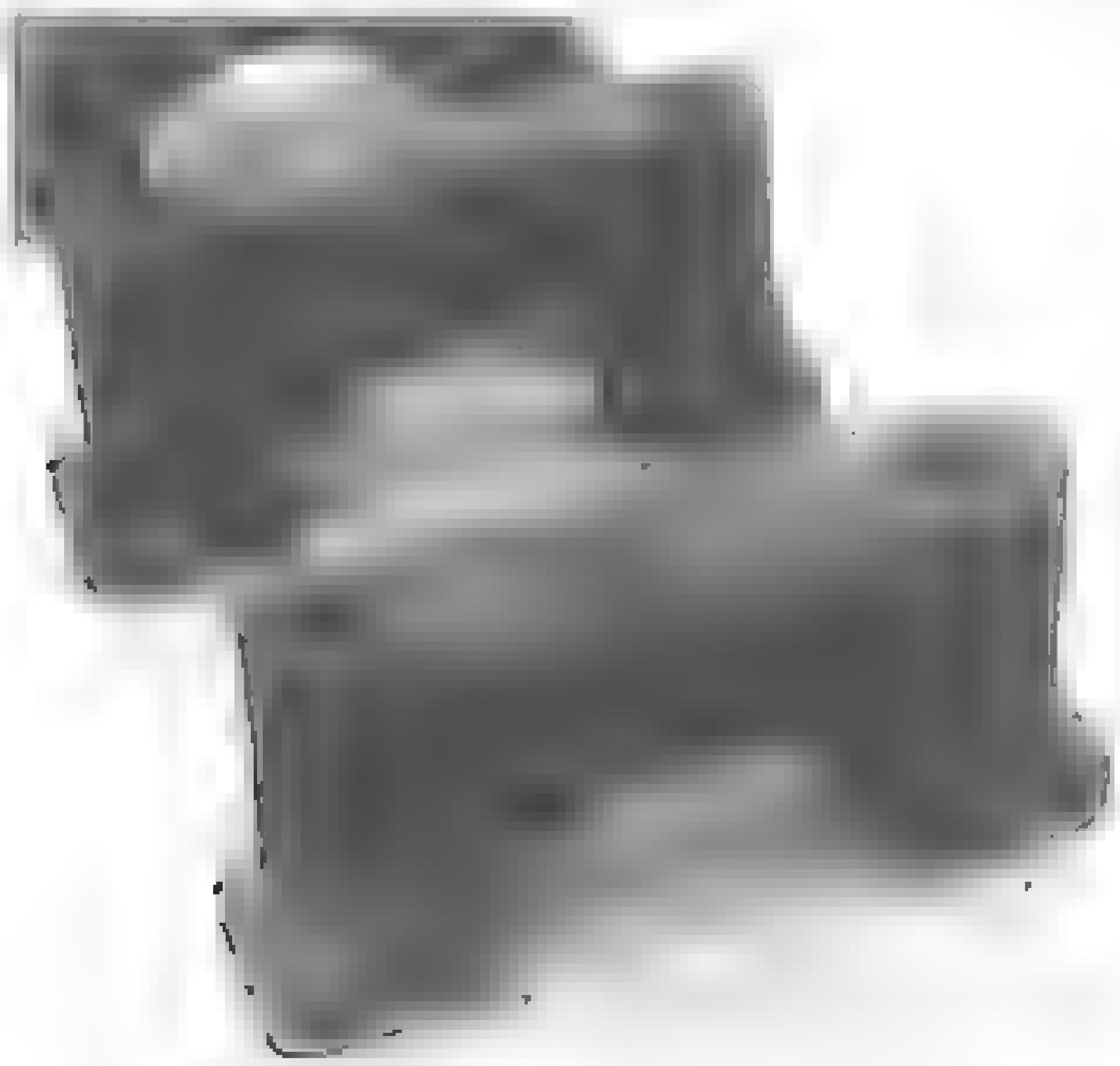
Finally, check the mating faces on the back and front of the block: they should be square to the crankshaft centre-line.

Two more inspections are necessary for racing engines. The camshaft bearing bores must be in line and each tappet bore must be of the correct diameter and perpendicular to the centreline of the camshaft. A tappet tipped off centre may easily dig into the lobe of a racing cam, causing premature wear and/or breakage.

About the only other parts of the block to which we have to give consideration are the main bearing caps and studs. I use ARP studs when these are available. Standard production main bearing studs should not be re-used on racing engines, and the same applies to big-end bolts. Usually, the standard cast-iron main bearing caps fitted to modern engines are acceptable but some older engines, or very-high-hp race engines, may require steel caps for reliability. In the very hottest motors I sometimes use a main bearing support saddle. This is a one-piece device that supports all the main bearing caps, or it may take the place of the individual main bearing caps. Either way it often extends out to the oil pan deck and transfers the bearing load so that it is shared by the outside of the block as well as the main bearing webs.

After all the machining work has been completed, the block should be thoroughly washed with hot, soapy water. Be sure to get all traces of honing grit scrubbed out of the cylinders, using a bristle scrubbing brush. Blow the block dry using compressed air, and spray all the cylinders, tappet bores and bearing bores with a water dispersant such as WD 40.

Stock square-shoulder main bearing caps like these are generally very st...



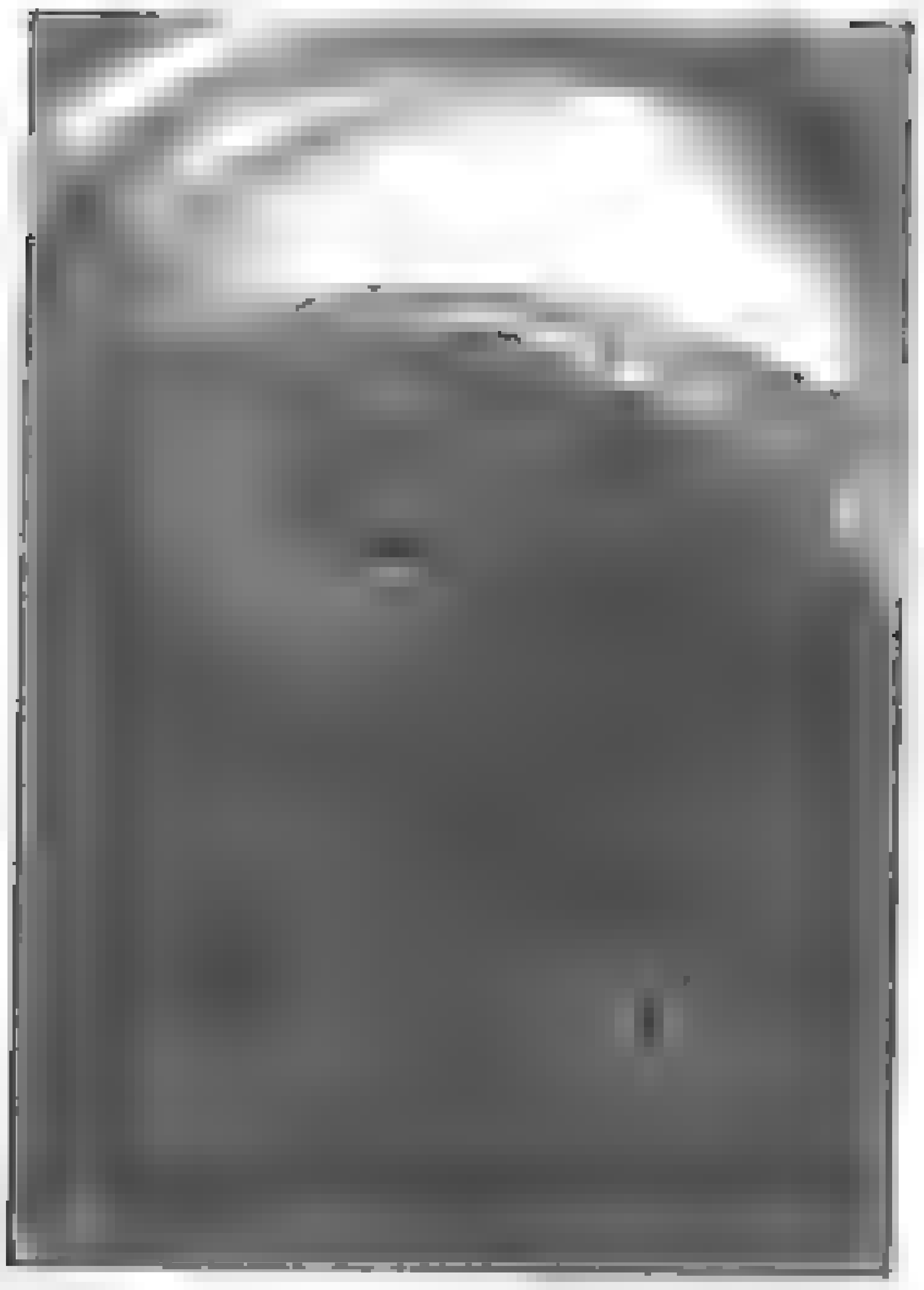
CRANKSHAFT SELECTION AND PREPARATION

With the block prepared we now turn our attention to the crankshaft. The majority of engines use cast nodular iron crankshafts, which generally give good service even in race engines if sensible rpm limits are observed and the rods and pistons are not too heavy. Some of these come from the factory with rolled fillets, which significantly increases their fatigue resistance and some may be heat-treated using a process such as Tufftriding. However, as the rpm limit is raised so the need for a better-quality forged steel crank becomes more apparent. In general the forged cranks fitted by Japanese and European car makers are very strong and suitable for high rpm competition use, some will also feature rolled fillets, and a few will be heat-treated by the nitriding process.

Not so good, however, are the forged steel crankshafts fitted by some American manufacturers. For example, the steel crank that Chev use in some of their small-block V8s is none too strong and cannot be recommended for use in competition engines: they are forged from fairly poor grade, and often 'dirty', 1053 steel. A nitrided version is also available, and while being a touch stronger they are production machined without suitable big fillets. In fact, Chev's cast-iron crank, if ground with decent fillets, will last longer in a race engine when limited to about 7,000rpm and less than 500hp.

Another worry with many production V8 cranks is the forging method employed. Typically they are forged 'flat', then the front and rear throws are 'twisted' 90° to give the required crankpin orientation. However, in production the throws will not be twisted exactly 90°, which means that not only the crankpins but also the counterweights will be 2 or 3° out. The raw forging has enough meat in it to enable the crankpins to be machined at 90° and the crank can still be balanced, but the incorrectly indexed counterweights will impose undesirably high bearing loads in a high-rpm competition engine. For this reason 'non-twisted' forged cranks, where all four rod journals are forged in place, are preferred for V8 race engines.

The sharp parting line on the web of this crank indicates that it is a cast-iron part



Left *This broad parting line ... rank. In a compe... ball that has been staked in ... cross-drilled ... replaced by a threaded ...*

Right *Rolled fillets improve ... subject to very high lo...*

Assuming that the standard crank ... must have it crack-tested ... any work on it. If it is free from flaws, carefully and fully demagnetise it ... same applies to rod ... attract ferrous metal particles ... area, which will soon wipe out the bearings. N... a check should be made to dete... ightness. Cranks can be str... generally it is a waste of t... es of combustion at ... the straightening process ... bent crankshaft increases bearing ... losses, it must be machined undersize by 0.010in or els... e than 0.002in.

Next measure each main bea... measure the diameter at several angles around ... end and in the middle of every journal, as wear can vary al... length. Ovality and taper should be less than 0.0003in in a high-perform... crank. If the wear is greater than this, grinding the journals unders... However, grinding weakens the crankshaft, so a crank replacement ... necessary in high-horsepower and high-rpm engines. When the cranks... ground, instruct the grinder to pay close attention to maintaining th... specified by the manufacturer (Figure 11.2); any reduction here wil... crankshaft. On the other hand, an increased radius serves to streng...

Figure 11.2 Austin Mini crankshaft

however, this should not be overdone or we can arrive at a situation where the big end or main bearings are locking on to the fillet. Narrower bearing shells, or bearings modified with a larger chamfer, can be fitted to avoid this problem.

Any casting slag should be ground off the crank and the area dressed. These slag spots can become a stress raiser or future failure point.

To assist in the smooth operation of the motor and to reduce bearing inertia loading, the crankshaft should be dynamically balanced; this will increase its life through a reduction in the shock loading and vibration that any imbalance would cause. Most manufacturers already do a reasonably good job of balancing their cranks, so unless the engine is going to spend a lot of time running in excess of 6,000rpm probably not worth the expense of having the crankshaft re-balanced. An exception to this is in the case of V8 engines. If these are fitted with pistons or rods much heavier than standard, the crank has to be re-balanced regardless of whether it is to be used in a road engine or a race engine.

Some manufacturers list Tuftrided replacement crankshafts, which are excellent for high performance road and rally engines. They can also be used in racing motors, but I would suggest that a strict eye be kept on rpm limits and that they be crack-tested each 300 miles.

THE TUFTRIDING PROCESS

Standard cast-iron cranks can be Tuftrided, but there is some risk involved. Any ferrous metal object can, in fact, be Tuftrided, but as the process requires immersion in a chemical bath at a temperature of 1,060°F for 180 minutes, internal stresses can lead to deformation. Cast-iron cranks are susceptible to bending and changes in journal diameter, so I feel that it is a better proposition to buy a pre-treated crank as they are

often not much more expensive than the standard item. However, do have the treated crank checked for straightness and phasing of the throws. Remember that cranks with rolled fillets should not be Tuftrided as this will destroy the pre-stressed condition of the fillets and weaken the crank.

There is a good deal of misunderstanding about the Tuftriding process, so I will [illegible]

The compound zone (0.0003–0.0005in deep in treated cast iron) is tough and [illegible] solution prevents incipient cracks from becoming fatigue failures, so the endurance limit of a cast-iron crankshaft can be increased by 20–60%.

HIGH PERFORMANCE CRANKSHAFTS

Forged cranks are more suitable for higher-output rally and racing engines. Forging increases the density of the component as the metal is literally squeezed into the required shape and compacted, resulting in a stronger core and better fatigue resistance. [illegible] engines.

In the UK competition cranks are often forged from EN40B steel, while in the [illegible] steel may be 'dirty', containing many impurities, or the alloy may not be up to specification. Also there is the question of what heat treatment the part has received after the forging and machining operations, as this has a significant influence on the [illegible] more strength and fatigue resistance, but because there may be a greater risk of failure [illegible] crankshaft manufacturer could choose a cheaper and 'safer', albeit inferior, heat treatment. Obviously such a crank will be less expensive to manufacture, but as it will have less fatigue resistance and a correspondingly shorter 'life', it may not be any cheaper in the long run.

Another factor influencing the cost of a racing crank is its weight. Naturally, complex machining costs money; a 20% weight reduction could add considerably to the price of the crankshaft. However, any reduction in reciprocating mass will pay dividends in improving acceleration out of corners, and in reducing bearing loads.

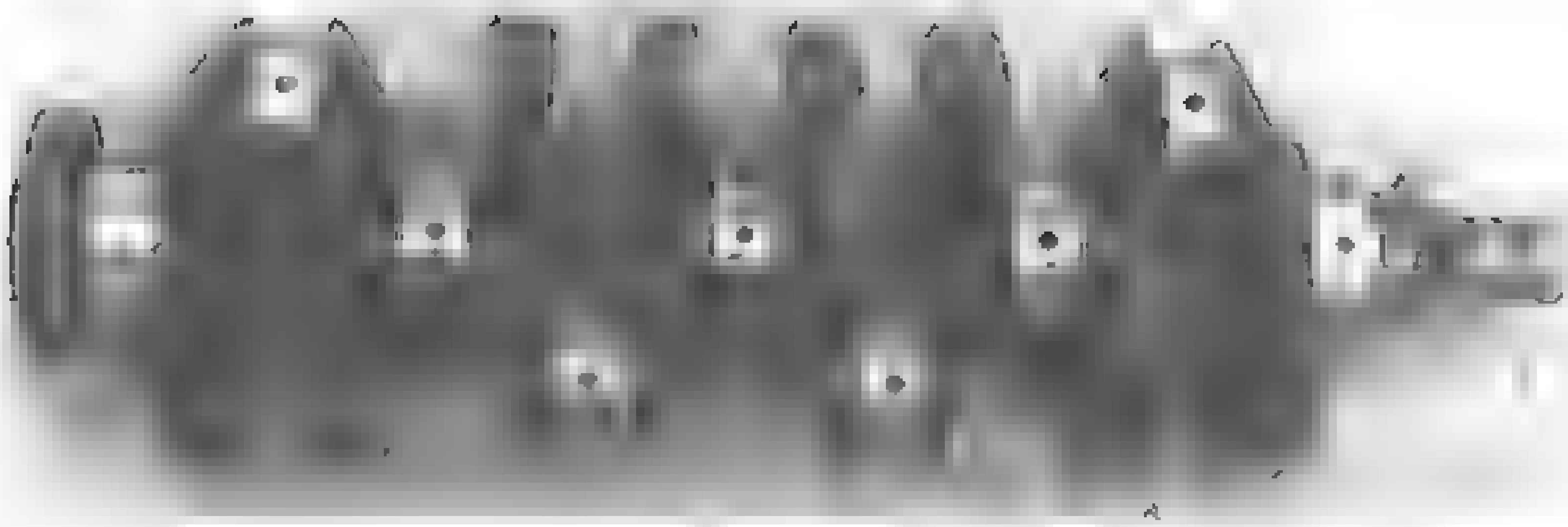
The most expensive, and strongest, cranks are fully machined from a bar (billet) [illegible] minimum then a 'cleaner' and thus stronger double vacuum re-melt steel, produced using vacuum electromagnetic induction melting, and costing around ten times as much as normal EN40B, will be used. After being machined the billet crank is heat treated to further enhance its strength and fatigue resistance.

The cranks of very-high-rpm or endurance race engines should generally be fully [illegible] As a result, the individual throws are subject to forces that, in combination, tend to make the crank whip and flex. Such increased loading at the bearings can wipe them out at high rpm, and in endurance engines fatigue failure of the crank could occur. The way to overcome these problems is therefore to counterbalance each throw. This does not change the overall crank balance, but achieves an internal balance for each throw. Obviously extra counterweights increase the crank's reciprocating mass and this in turn slows acceleration off the turns. Thus in sprint type engines, where bearings can be replaced frequently and the crank is regularly crack-tested, a fully counterbalanced crank may be a liability.

However, to prevent main bearing failure or case stretching in the VW engine, always fit a fully counterbalanced crank. Even in a very mild state of tune this is good insurance against engine damage. The standard crank will whip and pound the case, allowing the centre main to turn and wreck the engine. If you cannot afford a fully counterweighted crank for your air-cooled VW, have weights welded on to the standard crank.

In the high-rpm engine, crankshaft oilway modifications are undertaken to prevent bearing failure due to lack of lubricant. Many feel this means that engines such as American V8s, which have only the upper main bearing grooved, should have the main bearing journals cross-drilled. This is a total waste of money for road engines and it may not even be necessary for race engines. Looking at Figure 11.3, you will see what can be done to improve the quality of lubrication at the big ends while at the same time reducing the amount of oil that is pumped away from the main bearings. With this system, when one crankpin hole is blocked off due to centrifugal loads on the rod, the other hole feeds oil to the crankpin bearing. When carrying out this modification, great care is necessary to ensure that the new holes break fully into the original oilway.

Thus far we have only considered the crankshaft from the aspect of engine reliability, but it can also affect overall engine performance. Theoretically, crank throws should be of equal length and correctly indexed (phased). Slight differences in the stroke from cylinder to cylinder will not have a profound effect on performance, but incorrect phasing will knock power. A four-cylinder engine should have a piston at top dead centre every 180°, a six every 120°, and an eight every 90° (180° with a single plane crank). If any cylinder has a crank throw out of phase by 5° this will have the same effect on performance as an ignition or camshaft timing error of 5°. An expensive racing crank should have perfect phasing, but make a check to be sure.



A fully counterbalanced crank has improved f... e resistance and r... loads in a long-distance endurance engine, b... dditional rotat ng mass ... disadvantage, reducing accelerat...

Figure 11.3 Crankshaft oilway modification

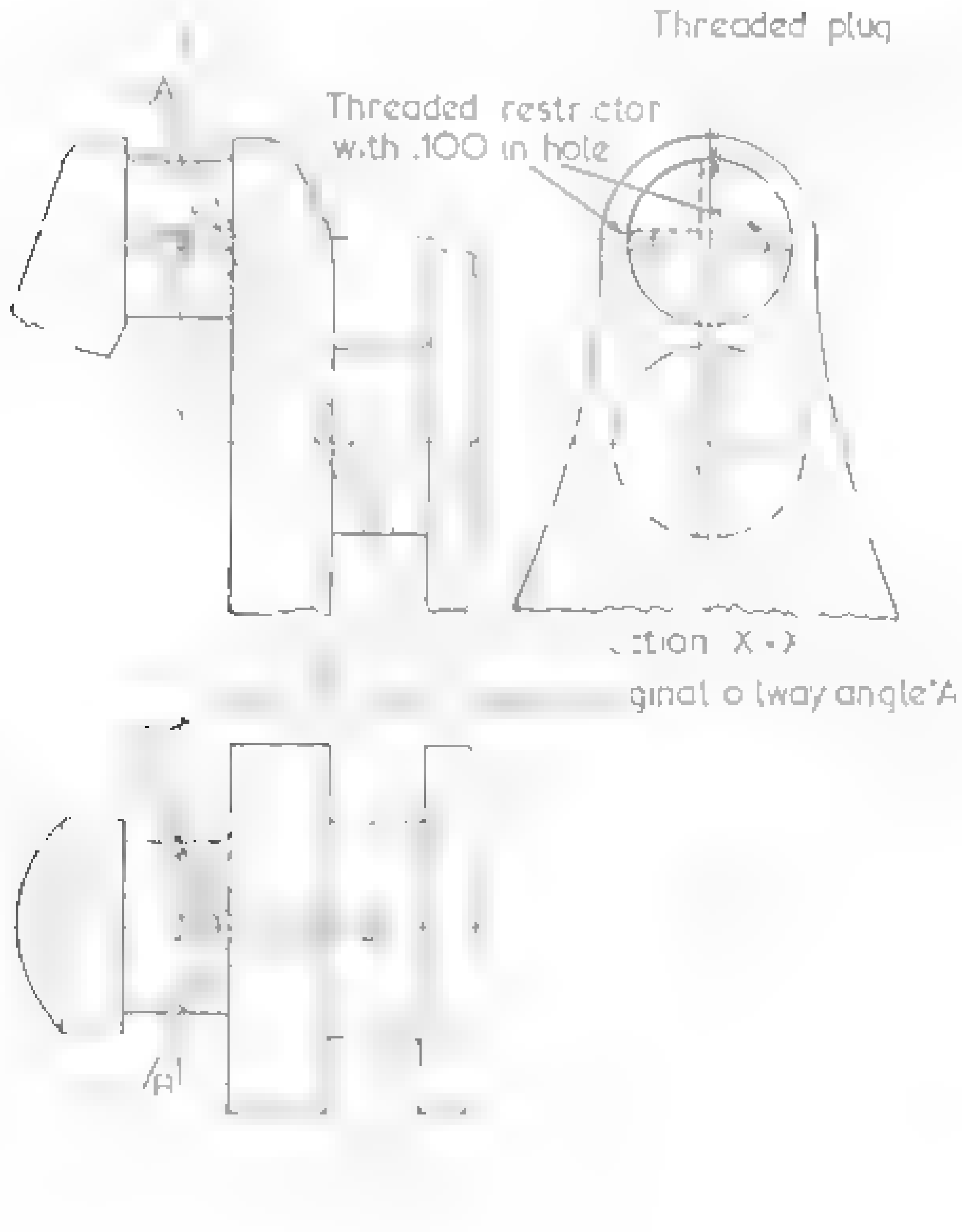


Figure 11.4 Crankshaft stroking by reducing the crankpin diameter

STROKER CRANK CONSIDERATIONS

[illegible]

Specially manufactured billet steel or forged cranks can be obtained with a [illegible]

[illegible] model vehicle by the same manufacturer

When a long stroke crank is fitted, some grinding may be necessary inside the [illegible]

CON ROD SELECTION AND MATERIALS

The connecting rod provides the mechanical link between piston and crankshaft, and is subjected to alternating compression and tensile loads. It is true to say that [illegible]

connecting rods in an engine have a tougher job to do than any other component, so it is not surprising that a large number of failures in high-performance and racing engines are caused by the rods letting go.

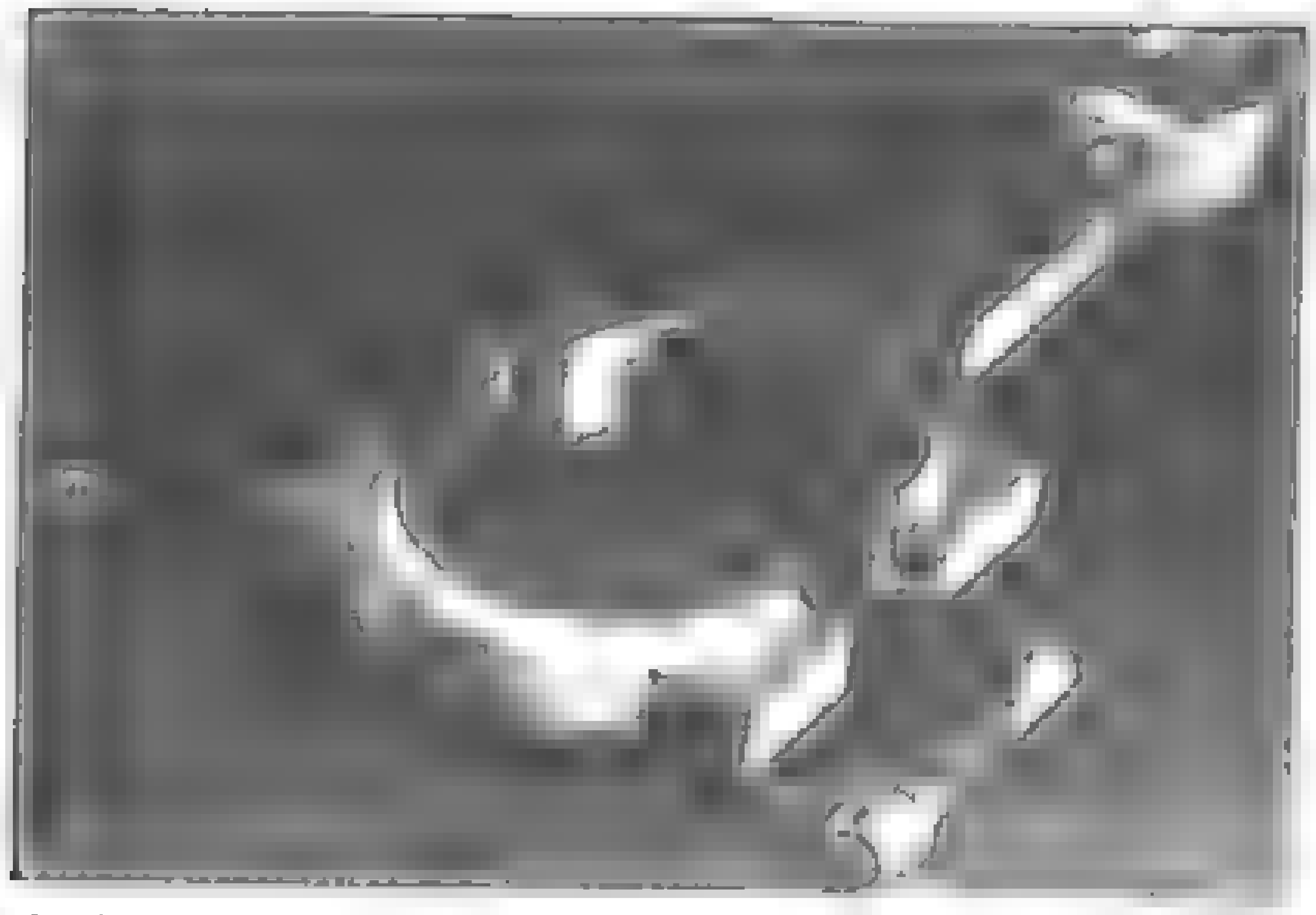
The highest load is reached when the piston is at TDC on the exhaust stroke. This tension load can range from a few thousand pounds in a small low-rev engine to well over 15,000lb in larger high-rev engines. Interestingly, this maximum load occurs on the non-firing stroke and is caused by the inertia of the reciprocating assembly comprising the small end, piston and pin. At TDC the piston is suddenly stopped, then reversed, and it is this sudden change that produces the high tensile load.

On the compression stroke the load is not as high (up to 11,500lb) as compression builds up slowly, and soon after TDC, when combustion has finished, the load changes from tension to a light compression load.

It is not only the fact that stresses are high in a rod, more importantly, they are applied and reversed each time the engine completes a cycle. This pulsing is much worse than if the loads were constant and applied continually, and it is this that leads to eventual fatigue and failure. A rod has to survive millions of stress cycles during a lifetime so it has to be tough and thoroughly prepared.

Most rods are forged from carbon steel or more recently powdered metal, but they may also be made of cast iron, aluminium or titanium. Cast-iron rods have no place in a modified engine and should not be used. Titanium is prohibitively expensive and is used only in big-budget race engines. Aluminium rods are light and reasonably strong, but fatigue quickly, so they should be used only in drag race and sprint-type speedway engines where frequent replacement is possible. Some time ago Chevrolet released some information on connecting rod fatigue testing that they had carried out. Some aluminium rods failed after 150,000 cycles. The heavy-duty small-block rod, polished and shot-peened, survived 1–2 million cycles at the equivalent of 7,500rpm, while the Chevy Bow Tie rod survived a minimum of 10 million cycles at 8,000rpm. When it is considered that the average race engine is required to stay together for 300 to 500 miles between rebuilds, the equivalent of 1–1½ million cycles, it is evident that care is needed, first in selecting the correct rod, and second in regular crack testing and replacement.

Powder forged rods are just beginning to gain limited acceptance among tuners even though some manufacturers such as BMW have used them in fairly potent engines, the quad cam V8s for example, since 1992. The powdered rod is a forging – its correct name is not a powdered rod, but rather a 'precision powder forged' rod, abbreviated to PPF within the industry. Like any other forged rod the PPF rod's suitability is dependent on its design shape, the metal used, heat treatment and final machining. I guess the thought that compressed powder can be strong doesn't line up with what we observe with wood products. A tree with a definite grain structure and interwoven fibres is comparable to a forged bar of steel. It is strong and will withstand massive forces for centuries, even millenniums in the case of some trees in California and Australia. When ground to dust though and compressed, the same wood is relatively weak and easily damaged. I guess in the back of our brain we expect powdered metal products to display similar weakness. However the powdered rod is not a simply compressed mass of iron filings, it is a precision, high grade product that can be suitable in a number of competition engine environments. It has several advantages over a conventional forged rod: it is cheaper for a rod of equal strength, it is lighter usually, because of the method of manufacture, tolerances can be extremely



This is a stock BMW V8 precision powder f

tightly controlled, so weights can be kept to ± 2gm, wit fractured big-end caps th cap is virtually locked in position on the rod which cuts problems with cap to rod movement and fretting, and helps maintain a round big-end bore which almost eliminates problems with bearings being pinched due to cap spread under load

The basic manufacturing process needs to be understood otherwise some tuner will continue to toss out stock rods that may be perfectly adequate for their perhaps even replacing them with inferior old-sty stock forged rods, others wil perhaps try to get by with weak old-style stock forged rods purely bec uninformed prejudice, when a set of strong PPF r were well within their budget

The base powder may be a high grade metal like 4340 (or 4260 which is superior). To this will be added special alloying materials and perhaps a lubricant such as graphite. This is all thoroughly blended and then fed to a compaction die where two punches p the powder into the shape of the die, in this case a con rod The next phase is sin ring – applying heat to the compacted powder to force the alloying formly throughout the rod, without actually melting. Next in a controlle atmosphere the rods are quickly heated to 825°C and forged to bring them to density They are then heat treated and finish machined

Connecting rods designed for the most stressful conditions are usu rged hined from 4340 or E4340 steel. However, as with crankshafts the of used is no guarantee of a quality rod. Many other factors such as steel purity, accuracy of machining, heat treatment, shot peening, design and weight also enter the equation

The standard rod is what we refer to as an 'I' beam rod because if you were to cut through the beam connecting the little and big ends the steel section would look like an 'I'. This style of rod can be very strong and manufacturers like Oliver, in particular Crower have proven that their 'I' beam rods hold up very well in high-rpm engines

ssibly the most respected manufacturer of con rods, Carrillo use an 'H' beam

Carrillo's superiority over other rods, at great expense I might add, is due to their tight control of the steel alloy (E4340) and similar tight adherence to their chosen heat treatment and shot-peening standards.

Many heavy-duty rods are more massive than the standard component, so there are a few things that must be considered during installation. As they are physically heavier, the crankshaft counterweighting will have to be re-balanced. Since the bi

block and camshaft

PREPARATION OF STOCK CON RODS

If you intend to retain the standard con rods, they must be carefully, and individually hand picked. Rods with any forging irregularities or indentations should be avoided. Lumps standing proud of the rod can be removed, but indentations are stress points likely to cause failure. The little-end eye should be in the centre with an even thickness of material around it. Again the big end should be well formed and symmetrical. Select rods that are reasonably equal in weight, a heavy rod may be excessively weakened if material was ground off to balance it with the lighter rods. If they are not already numbered, number each rod and cap, using a number punch, to ensure that they are never mismatched.

Standard connecting rod preparation must always include a check to ensure that the rod is not twisted or bent, as either condition will wreck the piston and or big-end bearings. Also necessary is re-sizing to bring the rods to an equal length. At the same time the big-end inside diameter should be checked for size and concentricity with the bearings fitted and the bolts torqued to specification. The little-end bush should be honed to give the correct piston pin fit.

The failure point for many rods is at the corner formed by the flat machined for the rod bolt and nut seat. These corners must be radiused to avoid stress concentration in this area. Every stock con rod will require some modification here (Figure 11.5).

These indentations are stress raisers, indicating that this stock rod should not be selected for use in a high output or high rev engine.

Figure 11.5 Connecting rod modification

ROD POLISHING AND SHOT-PEENING

The tough skin formed on the con rod by forging gives the rod much of its strength and fatigue resistance, so it should never be polished unless it is followed by shot peening, to create another work toughened and compressed skin. It is a waste of time polishing the entire rod, then having it shot-peened. If you look at the shank of a rod you will see, along its edges, a rough band where metal appears to have been sawn away. That is [illegible] as you can see. Of course, there is no hard skin along this ridge, in fact, its roughness is a stress raiser, so this ridge should be removed on a sanding belt. Give the entire beam a polish with fine emery cloth, then follow up with buffing and shot peening.

The shot-peening process has a very useful place in connecting rod preparation [illegible] under most forms of loading, maximum stress is at the surface, and second, surface [illegible] stresses at the surface will be reduced and any compressive stresses will be increased. However, because peening can seal over any surface cracks, all crack-testing must be done first. Also any straightening should be carried out before peening, or this will remove the [illegible] some growth and, if the peening has been done incorrectly, bending may occur.

CON ROD LENGTH

The relationship between con rod length and the crankshaft stroke, called the con rod ratio (con rod length ÷ crank stroke = con rod ratio), is of importance in the high-performance and racing engine. Most rod ratios range from 1.5 to 2.1, 1.65 to 1.75 is average.

For a long time it was argued that the length of the rod directly affected the performance of an engine. Basically the idea was that a long rod would improve top-end performance and also reduce piston and cylinder wear, while a short rod would pick up bottom-end performance, but with increased cylinder wall loads, piston and cylinder wear would increase.

Obviously a long rod will cause the piston to dwell at TDC longer and to move away from TDC more slowly, so there will be a change in the way combustion of the fuel/air mixture occurs. However, this does not affect the power output of an engine with flat-top, or fairly low dome, pistons. It is really only in race engines with very high top pistons, which badly upset the combustion process, that we see this influence coming through with an increase in hp in the upper rpm range.

With short rods the effect is reversed. The short rod accelerates the piston rapidly to TDC where it dwells for a much shorter time, before accelerating rapidly down the bore. When high-top pistons are fitted this rapid movement to, then away from, TDC tends to counteract to some extent the way the piston lumps disrupt combustion and the end result is that we see an increase in hp at lower engine rpm. However, again in engines running flat-top or low-dome pistons this influence is not seen.

This does not mean that if our engine has flat-top pistons we can forget about connecting rod length. With shorter rods, engine wear is a problem with which we have to contend in endurance racing. While in high-rpm sprint engines this may not be a concern, the increased cylinder wall and piston loadings evident in short rod engines certainly is, as any bowing of the cylinder reduces ring seal and the hp goes off. Hence I do not like rod ratios below about 1.65:1, and I would prefer it to be around 1.8:1. This is worth thinking about before you fit a long-stroke crank, as the tendency for the piston to rock in the bore is also increased as the rod-length-to-stroke ratio is reduced. Pistons with a reduced compression height and shorter skirt must be used with stroker cranks, otherwise the top would pop out of the cylinder and the skirt would contact the crank counterweights. This aggravates the wear and ring seal problem, as short pistons rock more than those of standard length.

CON ROD BOLTS AND CAP SCREWS

Rod cap bolts have to be designed to withstand a considerable force, to stop the cap from breaking away from the rod. At times it is necessary to fit bolts of a larger diameter than standard. Whatever the diameter, the tensile strength of the bolts should be 185,000psi minimum and it may be up to 270,000psi in extreme applications. However this is only part of the picture as to whether a particular bolt or cap screw will provide reliable long-term service in our competition engine. If it's an ARP (Automotive Racing Products) fastener you can be sure that it has been thoroughly tested to high loads for a million cycles, so will have excellent fatigue resistance. By comparison so called 'aircraft quality' fasteners are tested up to 45% load for only 100,000 cycles. What is worse for engine builders who think they are getting a better than normal bolt, because these have passed supposedly tough aeroplane standards, is that if all of the test samples endure only 85,000 cycles the entire batch is released for sale. In an average race engine that translates to a life of 25-45 miles. This isn't to imply that 'aircraft quality' bolts won't be reliable in your engine - they may be, but don't be lulled by those words into a false sense of security. Those magic words or

some fancy AMS (Aerospace Material Specification) identification should not be the basis on which your bolts are selected.

Prior to assembly, carefully inspect every big end bolt with about a 4X loupe ... ing for any nicks or imperfections ...

BEARING SELECTION AND CLEARANCE

... also have to absorb tremendous shock loads ...

Correctly tensioning rod bolts

... bolt ... bolt manufacturer. Do not ... tensioned ...

I use only Vandervell lead indium and Clevite 77 bearings because of their special features, both types being able to stand up to the worst competition loads when correctly installed. These bearings are steel backed, with a copper-lead intermediate layer that gives the bearing good fatigue strength and load-carrying capacity, as well as resistance to hydraulic break-out. The running surface for the Vandervell bearing is a precision overlay of lead-indium, while the Clevite 77 overlay is lead-tin.

Correct bearing clearance is an obvious necessity. Excessive clearance will promote knocking and pounding and allow excessive oil throw off into the cylinder bores; this in turn will cause higher frictional losses in the cylinder and increase oil consumption. Excessive clearance at the big ends will lead to oil starvation at the main bearings, with resultant bearing failure. Insufficient clearance will cause rapid bearing deterioration as a result of the increased temperatures that come about due to insufficient oil flow or a thin oil film (Table 11.3). A good rule of thumb is 0.0009in clearance for each inch of shaft diameter for main bearings and 0.0012in clearance per inch of shaft diameter for rod bearings when running 15W 50 fully synthetic oil at 100–110°C. With a thinner oil such as 0W-30, bearing clearances can be a little tighter. Con rod side clearance (end-float) also affects bearing lubrication, so this should be checked on every big end to ensure proper oil control.

Table 11.3 Bearing clearances for trimetal copper lead bearings

Diameter of shaft (in)	Clearance between shaft and bearing	Side clearance
1.5	0.0012–0.0017	0.004–0.006
2.0	0.0015–0.002	0.005–0.007
2.25	0.0018–0.0025	0.005–0.007
2.5	0.0022–0.0027	0.005–0.007
2.75	0.0024–0.0028	0.006–0.008
3.0	0.0025–0.0028	0.007–0.009
3.25	0.0025–0.003	0.007–0.009

Note: in engines where two con rods share a common crankpin (eg a V8) multiply the side clearance by 3 for steel rods, and by 4 for aluminium rods.

A slightly different situation exists when an aluminium alloy block is used in a race engine. Because aluminium expands more than cast iron when heated, what were ideal main bearing clearances at room temperature could become excessive at race engine temperature. For example, a clearance of 0.002in cold will increase to at least 0.003in, and more probably 0.004in, when hot. To address this situation the engine should be assembled with tighter main bearing clearances, which will frequently be close to what the manufacturer quotes for stock engines. Thus if the race clearance with an iron block is normally set at 0.002in, I would probably be looking at 0.001–0.0013in main bearing clearance when running an alloy block. However, with such tight clearances the oil and water should be pre-heated prior to engine fire-up.

MAIN BEARING INSTALLATION

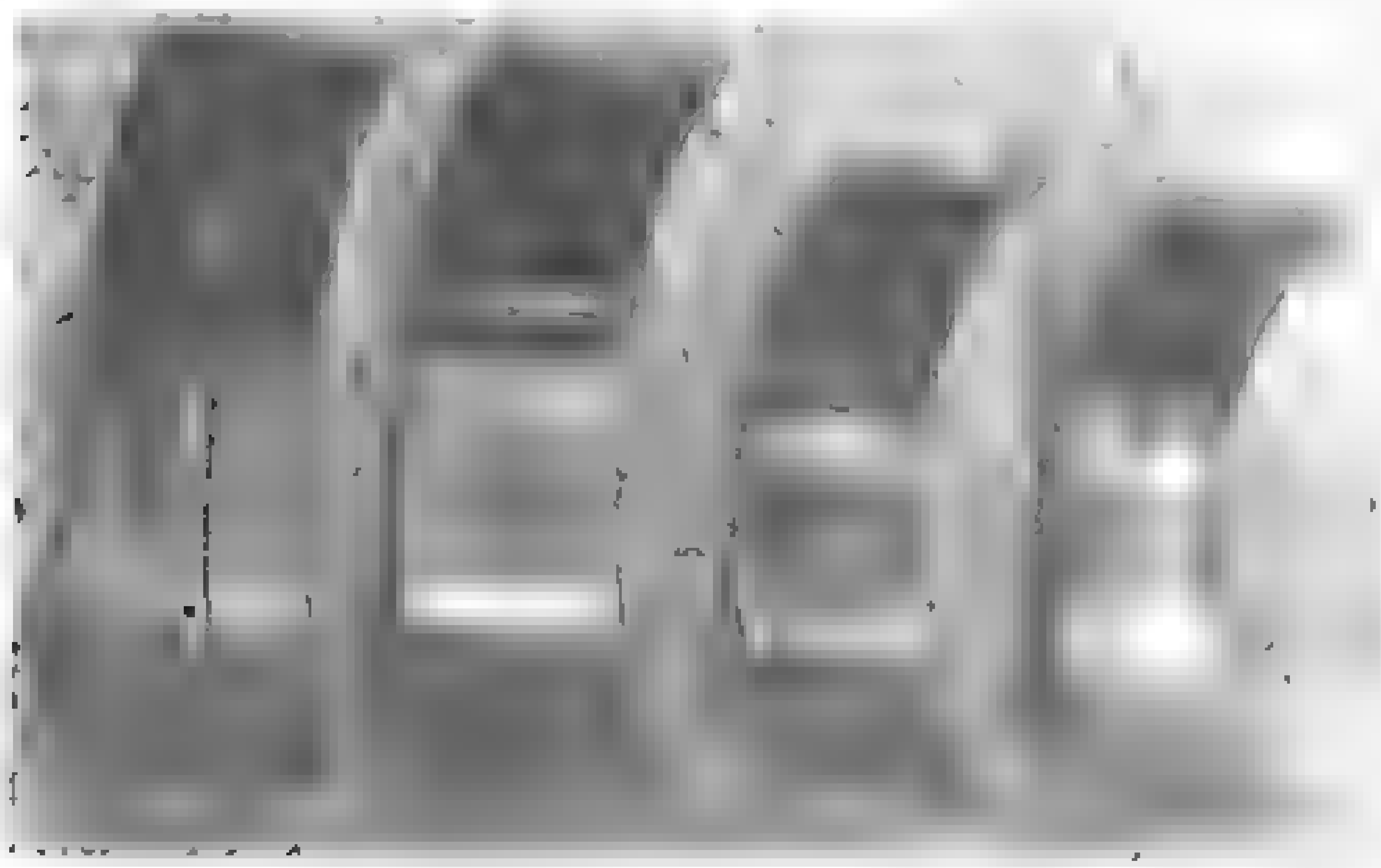
There are just a few simple rules to follow for correct bearing installation. Accurately measure and record the inside diameter of every main bearing and big end bore in

without bearings fitted. Now do the same with the crankshaft main and crankpin journals. Unwrap all the bearings and carefully wash them in clean solvent to remove the protective film, but do not polish them with any sort of abrasive pad as the overlay is very thin and is easily damaged. Do not worry about any little 'peaks' in the overlay – they will soon flatten out when the engine is run. Next measure the bearing shell thickness and double the measurement as there is a shell on each side of the shaft. The difference between the shaft and housing diameters and the doubled shell measurement is the space left for the running clearance. When measuring the bearing shell thickness be sure to take the measurement well away from the parting line as race bearings typically taper down by 0.0005in and up to 0.0015in over the last 1/4 to 1/2in. This eccentricity is manufactured into race bearings because at very high rpm the connecting rod bore pinches in due to rod stretch.

Many Clevite 77 and Vandervell bearings are available with three levels of eccentricity, 0.0005in, 0.001in and 0.0015in. Ideally a bearing should show wear over about 65–75% of its surface. If it is showing contact almost up to the parting line, fit bearings with more eccentricity, but when the contact area is too short bearings with reduced eccentricity are called for.

When you have calculated the running clearances for each bearing you will probably find that some have a little more than you want, while others may be a touch tight. If the engine is a popular unit for competition use the bearing manufacturer may have special 0.001in undersize and oversize bearings available. Using just one half of such a bearing allows the clearance to be opened up or closed down by just 0.0005in. When mixing a 0.001in undersize shell with a standard size shell the recommended orientation is to fit the shell with the thickest wall in the upper position in big ends and in the lower position if it is a main bearing.

Trimetal bearings have a very thin overlay that should not be polished with abrasive scouring pads. These big-end shells are identical, but the mains are not, the upper shell is grooved to provide oil feed to the big ends, while the lower bearing is ungrooved to increase its load-carrying capacity.



After ensuring that all the bearing housings and bearing shells are perfectly clean and dry, fit the shells and check oil hole alignment, any misalignment should be corrected using a small round key file. After filing, carefully dress the steel back of the shell to remove any metal fraze. Next coat all the bearings with engine oil and fit the crankshaft (do not use a 50–50 mixture of oil and any substance like STP). Fit the main bearing caps in their correct order with arrows facing the front of the block and gradually tighten them down. Before final tightening, the crank should be tapped to each side with a soft hammer in order to line up the bearing caps. Now check the crankshaft end float: it should be within 0.004–0.006in with a cast iron block. If the clearance is more than this, fit thicker thrust washers.

BIG END BEARING INSTALLATION

When it comes to fitting the big-end bearings follow the same procedure as for the fitting of the main bearings. There are still a few historic engines that use lock tabs on the big-end bolts. Throw these away, as they crush and give a false bolt tension reading. Use Loctite on the threads and you will have no problems with loose bolts.

Note that when the bearing shells are pushed into their respective housings they should feel 'springy', and 'snap' into place, this indicates good bearing crush. A lack of crush in new bearings indicates that either the bearing housing is distorted or that bearings with oversize shells are required to match housings that have been bored oversize. With used bearings a lack of crush usually indicates that the bearings have been hammered by detonation, but may also indicate a distorted bearing housing.

COATED BEARINGS

As well as going in for special oil additives, some tuners are advocating the use of products that must be baked on to bearings at very high temperatures for several hours, they are supposed to provide bearing protection when lubrication is inadequate. I do not recommend such products, as any time a bearing is heated in excess of 200°C for more than about an hour it will be ruined. This comes about due to the high temperature curing causing diffusion between the second and third layers of the bearing, which in turn weakens the overlay structure. Clevite 77 have produced a coated bearing, called TriArmor, using a lower temperature spray method to apply a 0.0003in polymer layer containing both molybdenum and graphite. This is intended to reduce friction, and in times of marginal lubrication when the bearing surface touches the crank scuffing is minimal.

PISTON SELECTION

Moving further up in the block, the next components for consideration are the pistons, piston rings and piston pin. First, you must decide whether the standard piston is up to the job. If the motor is in sports tune, it is quite probable that the standard cast and slotted pistons will be satisfactory. Higher states of tune will demand unslotted cast pistons, and in racing tune unslotted forged pistons will be necessary.

Most production engines use cast pistons, as they are easy to produce and shape as required. Some high performance engines will be fitted with a high quality forged or

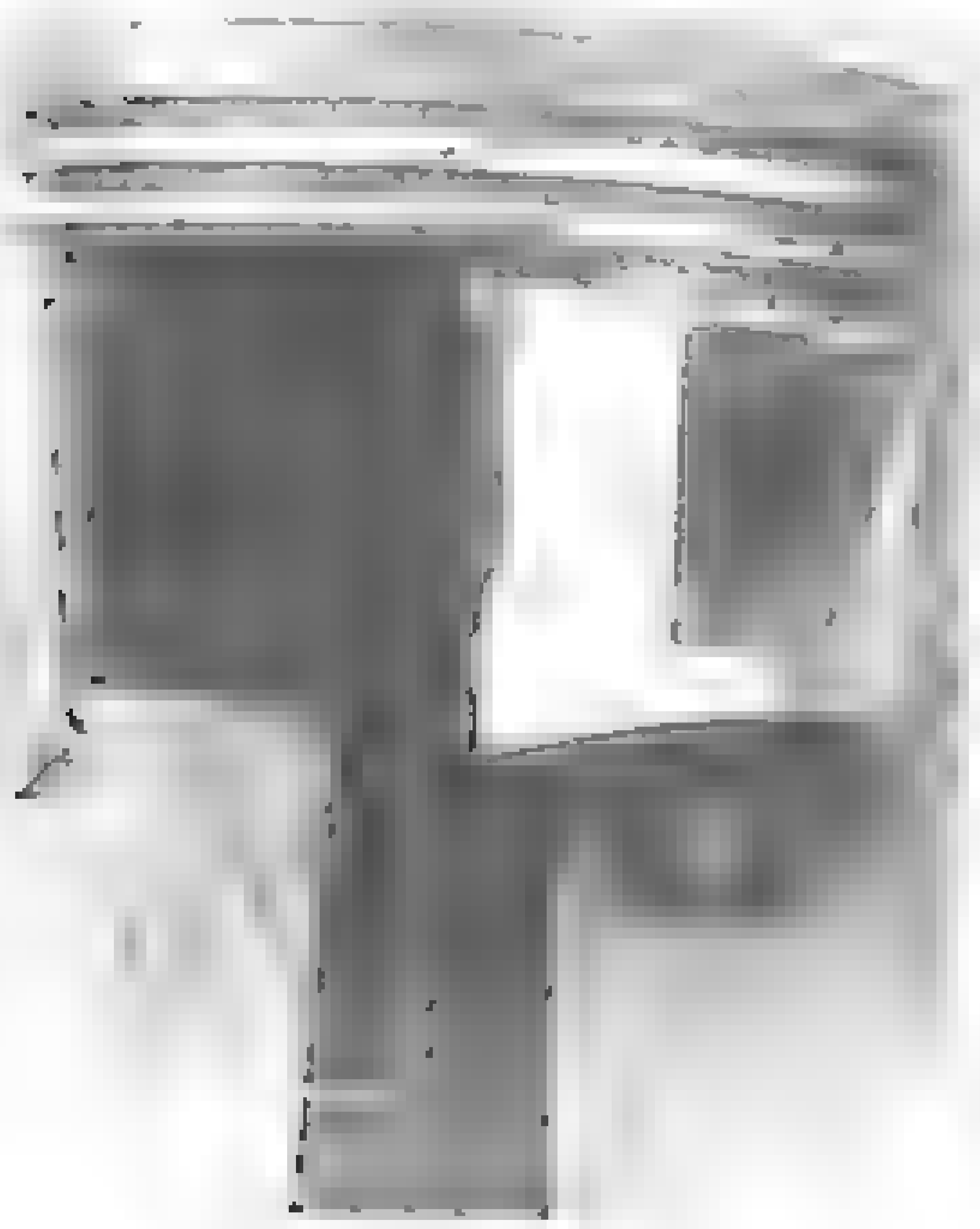
cast pistons right from the factory. By looking inside the piston you will be able to see whether it is cast or f [illegible] struts and braces.

The cast piston, either conventional or hypereutectic, has a relatively low [illegible] al density. Whereas the [illegible] performance road engines and budget race engines.

By comparison, forged pistons are much denser and consequently have a higher [illegible] strength. They are capable of [illegible] piston crown temperature reduction of 100°F is usual. Because of these beneficial features forged pistons are usually [illegible] short race engines.

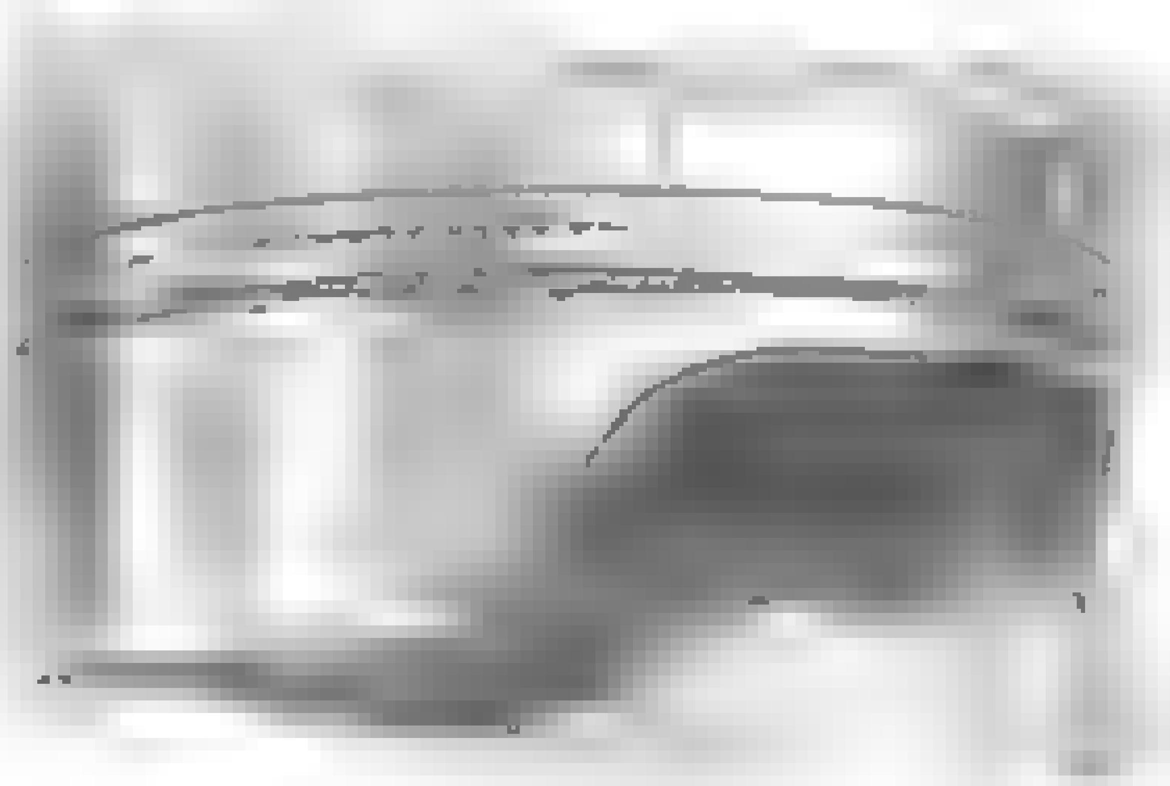
A less common piston, which some engine builders prefer, is the billet piston. [illegible] are fully machined from [illegible] recommend them.

The worst feature of many production pistons is the slot for oil drainage behind [illegible] l control ring. This [illegible] around to the other, on both sides of the piston. To make the situation worse, expansion control slots cut int [illegible]



Oil drain/expansion control slots seriously weaken the piston and make it prone to breakage in performance applications

A slipper piston has much of the skirt cut away and often has a very short gudgeon pin. This piston is for a 12,500rpm race engine, which is why it has only one compression ring



the piston skirt usually break into the oil drain slot. Slots weaken the piston considerably, allowing the skirt to break away from the top. In standard production engines this can be a problem, so you can imagine what you are up against if slotted pistons are retained in a high-performance unit. Pistons suitable for high-performance use (either cast or forged) do not have any slots for expansion control or oil drain-back. Instead, small oil drainage holes are drilled right the way around the oil control ring groove. Extra piston clearance and special design take care of expansion.

Racing pistons may be of either the full skirt or the slipper variety, and some may be a cross between both, with a skirt extending to below the pin boss, but cut back to the thrust faces below this point. The slipper designs are lighter but piston life can be as little as 250–300 miles at sustained high rpm. Also, depending on such things as rod length, crank stroke, piston diameter and piston clearances, piston ring sealing and oil control may not be as good as with a full skirt piston. Additional to these problems I sometimes find that I have to use a heavier gudgeon pin to reduce pin tower flex, and the heavier pin tends to offset the weight advantage of the slipper piston. Consequently I have a preference for the full skirt design. Together with a long taper-wall gudgeon pin (with a length at least 0.7 times the bore diameter), piston life will be anything from three to six times longer than an equivalent slipper type.

RING GROOVE ACCURACY

Apart from durability the next most important thing I look for in a piston is accurate machining of the ring grooves. Twenty years ago it wasn't such a big issue as ring manufacturers were struggling to supply us the best competition rings parallel and flat to within ±0.0005in. Today the flatness and parallelism of the best quality competition rings can be held to ±0.00005in; a tenfold improvement, fortunately there has not been a similar jump in price, but at this standard they are certainly not cheap! To take advantage of this advance, and avoid wasting money on accurate but expensive rings, we need to find pistons with a matching level of accuracy in the ring grooves. This has to be true of the piston not only when new out of the box, but also well into its service life. Obviously there is little to be gained if, because of metallurgical or structural design problems, the ring lands move soon after the you begin to work the engine hard. Mahle for example can supply their top of the line pistons with land flatness held to within a micron (0.00004in). Such pistons cost two to three times that of ordinary off-the-shelf forged competition pistons. In absolute peak hp terms such pistons and a matching ring package will be worth 1–1.5% on a newly built engine. As the engine nears the end of its service the gain will be about double those numbers, and lower in the rev range the improvement may be a bit higher.

We have always known that flat lands and flat rings seal better as the piston rises on the compression stroke, and then descends on the power stroke, by blocking gas from around the back of the ring escaping between the bottom of the ring and the land. Therefore it was always considered vital that the bottom of the ring and the lower side of the land never be scratched or damaged in any way. However in reality the same is true when it comes to the top of the ring and the top of the land. The seal here is just as important if we are going to make good power. We have to remember that in a naturally aspirated engine we rely solely on the vacuum created within the cylinder as the piston descends to draw in the air fuel mixture. The bigger the vacuum the more

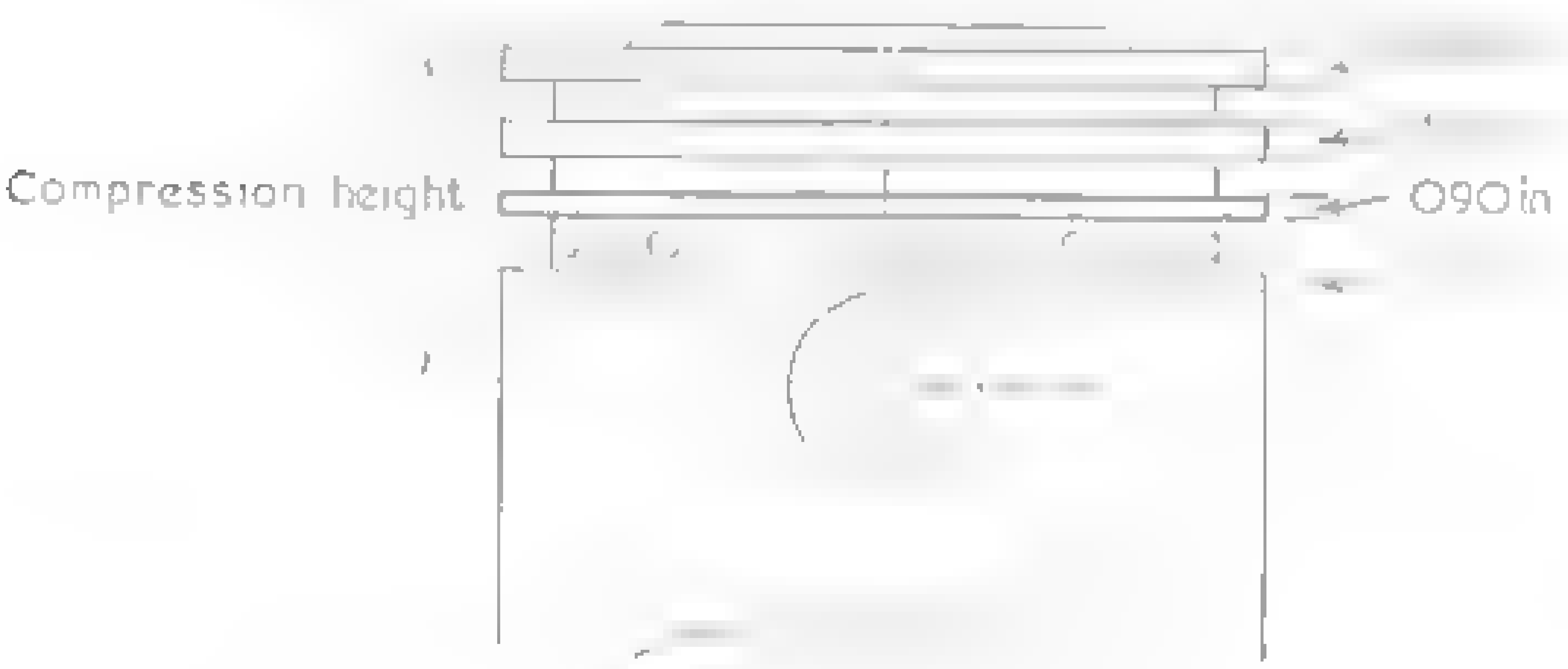
power producing mixture we draw in. Therefore the better the rings seal, not only on the power stroke but also on the induction stroke when the vacuum is being developed, the more hp the engine will deliver. The only way we can pull the best vacuum is by using a piston/ring package that will seal efficiently on the down stroke. This means that the flatness of the top of the ring groove, and the top of the ring, is equally as [illegible] side seal must not be compromised in any way. That means being careful not to scratch or nick any side of the ring groove or any side of the ring. It also means that a piston should not be machined anywhere after the grooves have been machined. Put another [illegible] on a piston should be the machining of the ring grooves.

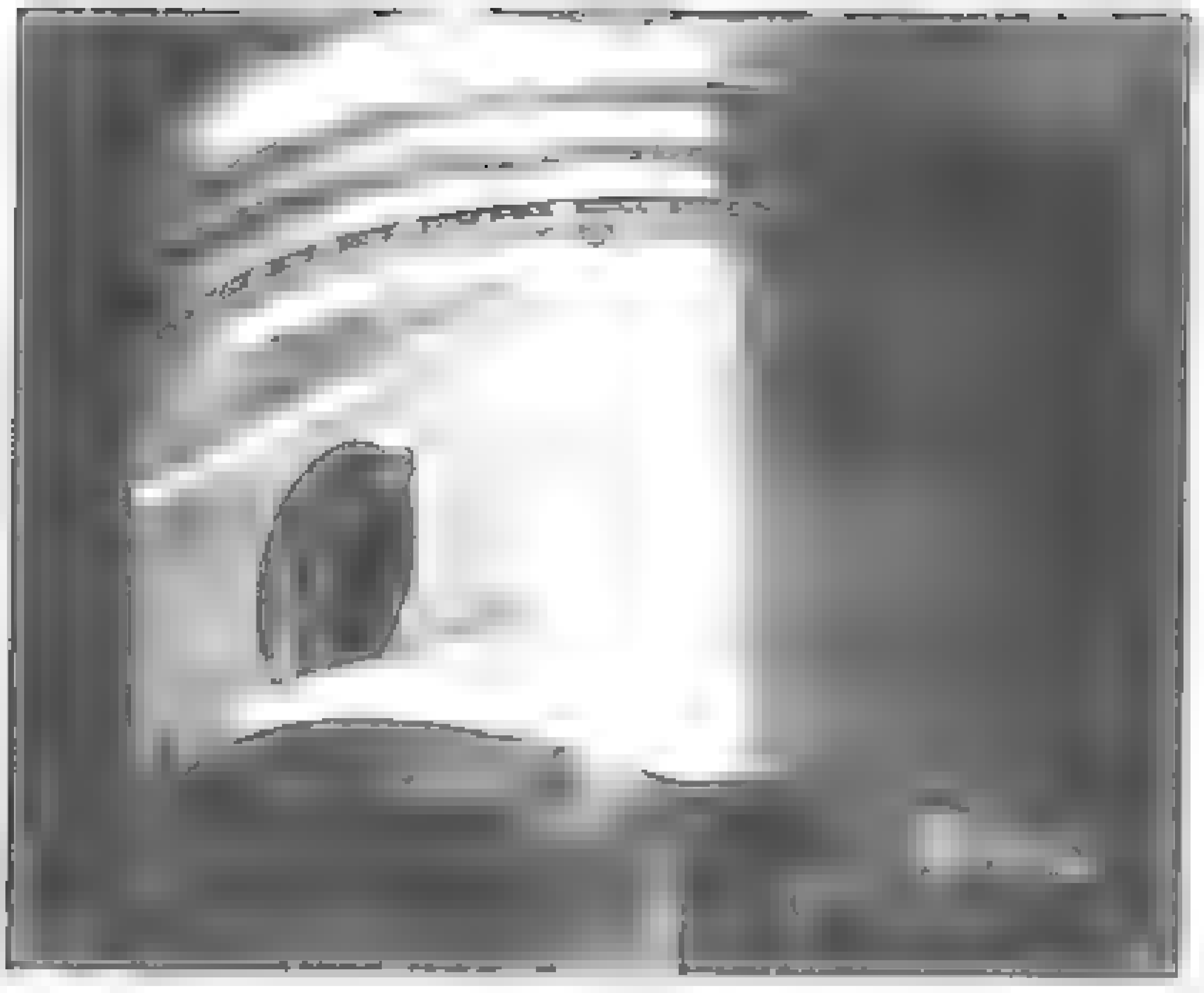
RING GROOVE PLACEMENT

The actual placement of the rings is important to gain those last few hp. As shown in Figure 11.6 we want the rings as high up the piston as possible, but this is limited by the engine's hp potential and the heat load on the top ring. This means that in engines producing up to about 65hp per cylinder the top ring should be 0.180in from the crown and the second ring should be 0.0160in below the top ring, with the oil control ring a further 0.090in down; while at higher power levels the top ring will be at 0.240–0.300in. High ring placement brings a number of benefits. First, we can locate the piston pin higher in the piston which permits the use of longer con rods, which in turn lowers the piston skirt loadings, so there is less risk of scuffing and piston distortion; they also reduce piston rock at TDC, so ring seal is better and hp rises. Second, high placement of rings allows the piston crown to shed its heat load more rapidly and with the second ring also placed high it is able to play a much greater role in getting heat out of the top of the piston, rather than leaving this task to the top ring. In turn, with a greater burden of heat transfer being shared by the second ring it is now possible to reduce the width of the top ring, the benefits of which we will cover later.

Rapidly ridding the piston crown of heat increases piston durability, and a cooler piston top lessens the risk of detonation as the incoming fuel/air charge will not be

Figure 11.6 Piston ring placement





This full skirt piston has been machined across the pin boss area to reduce its weight and cut friction, and to allow the use of a shorter and lighter pin. Note the 'H'-section con ro

preheated to such an extent. This opens up the possibility of greater engine reliability with the engine being able to run hard[illegible]r longer in endurance events. Alternativel[illegible] we may choose to sacrifice some reliability for more hp and so opt to bump [illegible] the compression ratio or the spark, or else lean the mixture off closer to full power lean.

PISTON-TO-CYLINDER CLEARANCE

We tend to think of pistons as being round, but actually the skirt is cam-ground an oval shape, and the piston also tapers from bottom to top, both features be[illegible] necessary to prevent seizure. The top of the piston is almost twice as hot as the bottom of the skirt so it expands more. Due to the extra metal around the pin bosses, more heat is directed to this area, elongating the piston a[illegible] the piston pin axis. To compens[illegible] skirt is ground oval. Most pistons are from 0.005–0.012in less in diam[illegible] pin axis than across the thrust f[illegible]. Be careful to measure piston clearance only [illegible] the thrust faces, and either at the bottom of t[illegible] skirt or up near the pin (check this with the piston manufacturer).

Accidentally dropping a piston may damage the skirt and lead to eventual seize due to skirt distortion. Never bang a piston pin out with a hammer and drift [illegible] th[illegible] way to push the skirt out of shape. If the pin will not push out easily, heat th[illegible] piston in boiling water or oil, then gently tap it out with the con rod secured in a vice to prevent any pressure at all on the pisto[illegible]

For better lubricatio[illegible] to allow for extr[illegible] [illegible] expansion, high p[illegible] [illegible] must have more p[illegible] [illegible] than mode[illegible] [illegible] [illegible] best to consult [illegible] [illegible]s, but in general I w[illegible] looking at a minimum of 0.002in [illegible] up around 0.003 [illegible]

engines running cast pistons (including hypereutectic cast) with bore sizes in the 80–110mm range. In a race engine cast pistons would require a little more clearance 0.0027in–0.004in [illegible]

VALVE-TO-PISTON CLEARANCE

As the majority of motors have heads with inclined or canted valves, some thought must be given to providing adequate valve-to-piston clearance. There should be a minimum of 0.060in vertical clearance, although for push rod engines, which in competition are more likely to suffer valve float, I prefer 0.080in for the inlets and 0.100in for the exhaust. The cut out diameter for safety is 0.120in greater than the valve head diameter, but in race engines, where there is no build-up of carbon on the valves or pistons, this can be reduced to 0.050in.

There are two basic ways to go about checking the safe valve working clearance, [illegible] and the head and head gasket need to be fitted.

The first way involves the use of a piece of modelling clay pressed down on the top of a piston. After the clay has been smoothed out, spray it and the valves in the [illegible] This will help prevent the clay from sticking to the valves later, when the clearance check is made. Fit the [illegible] valve cut-outs. Next cut the clay, using a sharp, wet knife, to check the clearance around the periphery of the valve head.

The other method is accurate only if the valve cut-outs are central under the valve, as there is no provision for a check to be made on the clearance between the edge of the valve head and the edge of the cut-out. In this instance light checking springs need to be fitted instead of valve springs. First tension down the head and head gasket as before, and adjust the valve clearances. Next set up a dial gauge on the inlet [illegible] the dial gauge and push the valve open as far as it will go. The dial gauge reading gives the clearance. Measure the exhaust valve clearance similarly.

PISTON-TO-HEAD CLEARANCE

To assist in equalising compression and combustion pressures, the crown of every piston must rise to the same point in each cylinder; this is called the deck height of the

piston. Generally we would want to be running minimum deck clearance to assist cylinder scavenging, and to increase the compression ratio. A good solid motor with steel rods can run with a minimum of 0.040in clearance between the piston crown (not the compression lump) and the squish area of the head. Therefore if the head gasket is 0.030in thick when compressed, we could run a minimum deck clearance of 0.040 – 0.030 = 0.010in. This means that the pistons would all be machined so that they are 0.010in from the top of the block at TDC. If one piston is 0.015in down the bore, it will be necessary to machine all the other pistons so that they are also 0.015in from the block deck at TDC. To restore the compression we would then need to machine 0.015 – 0.010 = 0.005in from the block deck.

Aluminium connecting rods stretch more at high revs, and aluminium has a greater coefficient of expansion than steel, so we would need at least 0.070in piston-to-head clearance. Engines that have steel rods but suffer block flex or crankshaft whip, eg air-cooled VW, will require 0.060in clearance. Most American V8s are more rigid and can run 0.045in given the full treatment of steel crank and rods and heavy-duty block. If a standard block and crank are retained, increase the clearance to 0.050in.

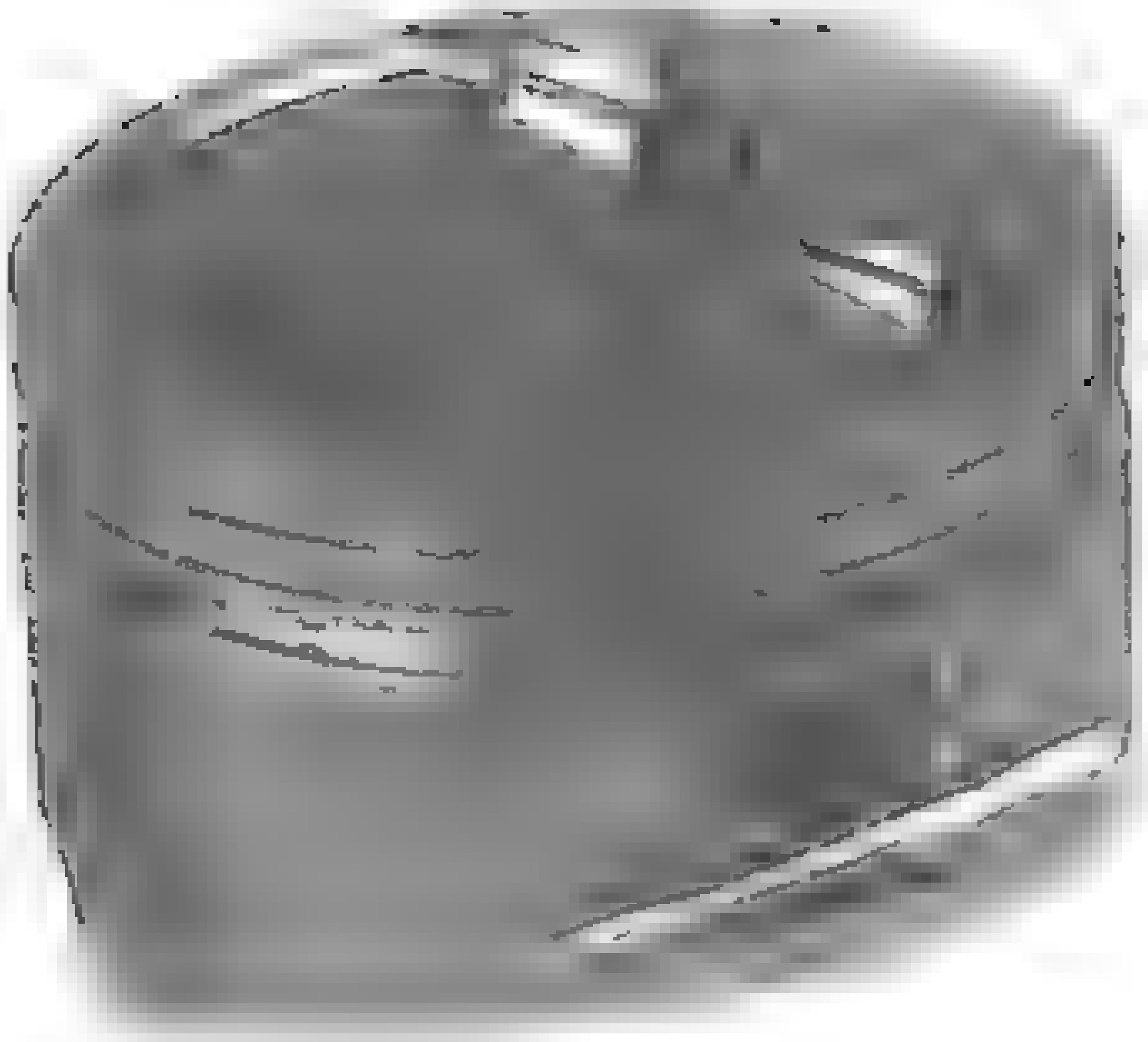
DECK CLEARANCE AND SQUISH IN ROAD ENGINES

In road engines I like to keep the squish clearance between 0.035 and a maximum of 0.045in. Usually, to achieve this figure the top of the block will have to be milled. Note in a road engine that it is not essential for all pistons to rise to the same height in each cylinder; a variation of 0.005in is acceptable. The reason I like to build a good deal of squish into a road engine is this causes the piston to squish the fuel/air charge from the edges of the cylinder toward the spark plug. The fast moving gases meet the spark plug and quickly carry the combustion flame to the extremity of the combustion chamber, thus reducing the risk of detonation. This opens the way to use a higher compression ratio to enhance part throttle performance and fuel economy.

With the passing of time more benefits of building squish into the engine have come to light. The mixture being purged across the combustion chamber from the squish areas homogenises the fuel/air mixture more thoroughly and also mixes any residual exhaust gas still present with the fuel charge. This serves to speed up combustion by preventing stale gas pockets from forming. Such pockets slow down and in some instances can prevent, flame propagation. With improvements in combustion quality, performance is improved particularly at cruise. Consequently the engine is markedly more responsive, plus fuel consumption falls.

Turbulence caused by the squish effect also serves to enhance heat transfer at the spark-initiated flame front. Without proper heat transfer, jets of flame would tend to shoot out toward the edges of the combustion chamber, prematurely heating the surrounding gases to start off the cycle leading to detonation.

Whenever dished or reverse dome pistons are fitted to lower the compression, the outline of the dish should ideally be a mirror image of the outline of the combustion chamber. This means that the piston has a squish surface which exactly matches the cylinder head squish area. Unfortunately, pistons machined to mirror the combustion chamber tend to be much more expensive because of the intricate machine operations involved. Hence, car manufacturers tend to use lower cost pistons with a central dish and narrow squish band or an offset dish which places a large squish area



This piston has a squish area that matches the cylinder head squish area

over to one side of the piston crown. When the piston squish band is as wide as the cylinder head area the required squish is produced. However, the squish band on some pistons is much narrower than the cylinder head squish area, and some tuners when lowering the compression ratio also machine the squish band too narrow

SQUISH IN RACE ENGINES

The situation is different with competition engines. Here low speed combustion quality and cruise economy are not usually a concern. What we are after is the best hp over a range of about 35–45% engine revs at the top of the power band, usually on throttle openings wider than 60% open Under such circumstances mixture density is quite high so there is a corresponding increase in mixture turbulence as the piston rises to TDC on the compression stroke. Therefore the squish area can be reduced or the squish clearance can be increased in many race engines without any power loss Perhaps there may be a power gain

I have experienced engines on the dyno that sounded good and sharp with wonderful throttle response, but with only average power output. Frequently opening up the squish clearance would see a nice power gain and much smoother running The conclusion I reached was that tight squish was causing some sort of combustion instability, or possibly the very rapid compression as the piston squeezed the gases between the squish faces meant that the gases became the equivalent of solid matter so in effect it was like the piston banging into the head

While you digest that last thought keep in mind that engines literally can go solid or 'hydraulic'. The problem is more often noted in nitro burning engines running pistons that are not dished, but I have also seen it in engines operating on very rich alcohol mixtures and also when crude water injection systems have been employed What happens is that as the piston accelerates up the cylinder it collects all the

unvaporised fuel or water. Violent wind across the piston crown disperses this liquid away from the centre of the piston out toward the edges. If the clearance between piston and head is sufficient no harm is done as the piston comes up to top dead centre. If it is too close the engine goes solid as the piston attempts to compress the liquid against the head. There is no time for this liquid to be gently squeezed out of the squish gap into the combustion chamber as some suppose. Nor does fuel or water puddled into one area on the piston crown get 'flattened' or dispersed across the entire squish area. Basically the engine damage will range from distorted pistons through to a totally destroyed engine.

In two stroke engines the squish areas are extremely tight, down to 0.020in, but best combustion often results from squish areas that diverge toward the centre of the combustion chamber. Thus, while the squish clearance may be only 0.025in at the very edge of the piston, it may open up to 0.032in about 16mm from the edge. I have experimented with the same idea on four strokes and while it had the desired effect it did not appear to be superior to simply opening up the squish clearance a little, or reducing the squish area.

PISTON CROWN MODIFICATIONS TO AID COMBUSTION

As I mentioned right at the beginning of the book, the top of the piston is a part of the combustion chamber, and those high compression lumps on the top of your pistons can [illegible] may retard flame travel after ignition and upset complete cylinder scavenging during the exhaust cycle. On the intake stroke, the dome can restrict the critical initial air/fuel flow, then disrupt final mixture homogenisation before ignition.

To improve air flow into the cylinder at low valve lift, the intake valve pocket must be laid back to reduce valve shrouding (Figure 11.7). As the initial flow is along the floor of the port and around the short radius toward the squish area of the head, this modification will assist in cylinder filling. Many tuners worry about removing metal from the piston dome because of the slight drop in compression ratio; however, the improved flow and better combustion will far exceed the slight power losses due to the drop in compression ratio.

After ignition, we want the flame to travel smoothly across the piston dome and back towards the squish area. Any abrupt edge on the compression lump will disrupt the flame front as it moves across the piston, leaving pockets of mixture that are either unburned or only partially ignited. Rounding and smoothing the top and sides of the dome does much to reduce this problem.

The exhaust valve cut-out must not be laid back, as this is usually not advantageous. The sharpness of the exhaust valve pocket does cause some flame disruption, but exhaust flow and cylinder scavenging are usually superior when the exhaust valve is shrouded by the cut-out.

After the engine has been run for a time, a careful study should be made of piston dome coloration, which is a surprisingly accurate means of investigating combustion completeness. Keep in mind that combustion begins before the piston reaches TDC and continues well past TDC. If at any time something happens to cause the flame to stop, the piston colour will indicate where the dome is causing the cessation.

All motors with bath tub or wedge-chamber heads will show an area of no carbon build up on both the piston and cylinder head. This is in the squish (or quench)

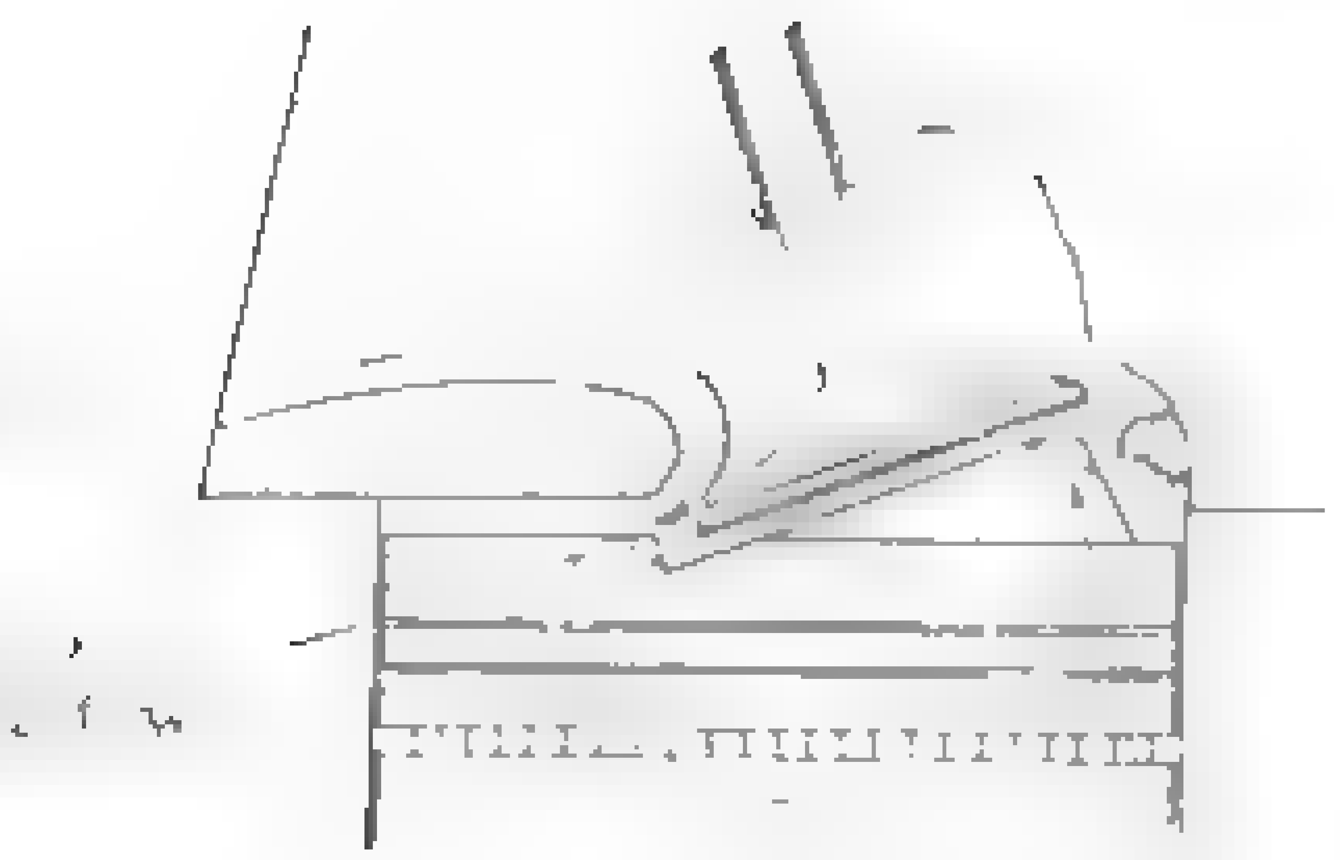


Figure 11.7 Piston dome modification.

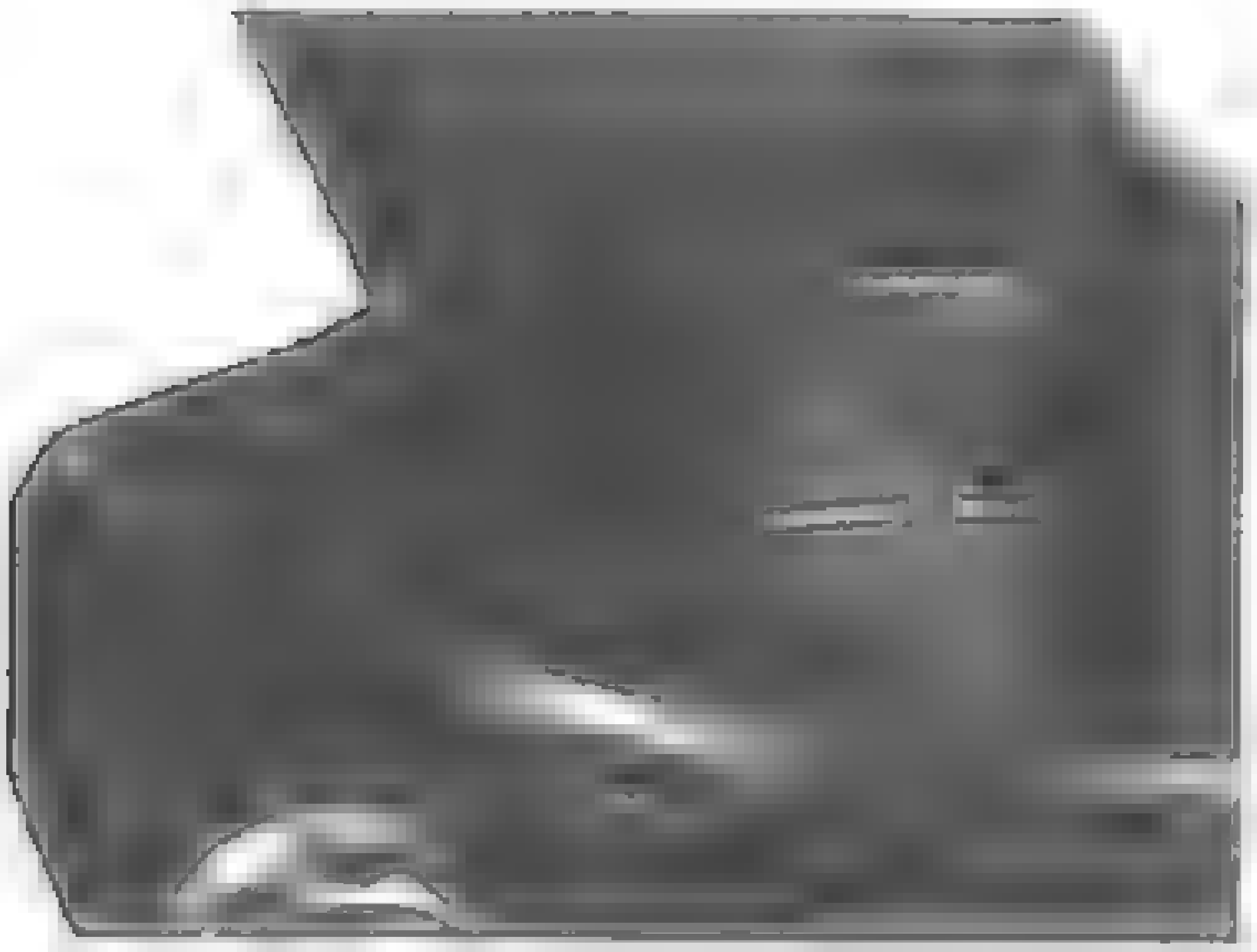
area where no combustion takes place due to the closeness of the flat face of the head and piston. Any other areas on the piston dome where there is little or no carbon build up likewise indicate that the dome is coming too close to the combustion chamber and is retarding the ignition flame at and around TDC.

The solution is to machine the compression domes lightly until the carbon build-up is more even in quantity and colour. Because the flame moves from the spark plug out across the combustion chamber, you will need to keep in mind the direction of flame travel when you come to remove any metal from the dome. Generally, it will not be necessary to machine the entire uncoloured area as it is usually the area immediately in front of and around the beginning of the uncoloured area, relative to the direction of flame travel, where the dome is coming too close to the combustion chamber wall and creating a pocket of unburned fuel. The place of least flame activity is usually around the high side of the piston dome in the area of the intake valve cut-out, close to the cylinder wall. If you can modify the dome to colour evenly in this area, you are well on the way to achieving good combustion.

PISTON PIN AND PIN CLEARANCES

As mentioned previously, I prefer a long piston pin with a tapered bore to cut its weight. However, if we go too far we end up with a pin that is nice and light but which allows the piston to flex excessively. The end result will be cracking in the top of the piston where the pin towers join the crown. The ends of the pin should be chamfered to bear up into round wire circlips and the length should be such that when the pin retainers are fitted, pin end float is preferably 0–0.001in; more than 0.005in is unacceptable in a race engine.

When we add vacuum to the crankcase to reduce windage losses we begin to experience marginal lubrication in a number of areas, the piston pin being one such component. The fix may take several forms including a squirter nozzle in each cylinder to spray oil up into the piston crown/pin area to both lubricate the pin and cool the piston, or con rods with an oilway from the big end up to the little end to lubricate the pin. Another route is to fit pins with a Diamond Like Coating (DLC)



[illegible] nuous oil flow up the rod to the little-[illegible]

As well as enabling the pin, piston and little end to perform very well with only l[illegible] lubrication, a pin with a proven DLC such as Casidiam from Anotech enables us to run rods with an unbushed little end. This then translates to a smaller, lighter [illegible] reduces bearing loads and may enable us to cut bearing size, reduce crank weight et[illegible]

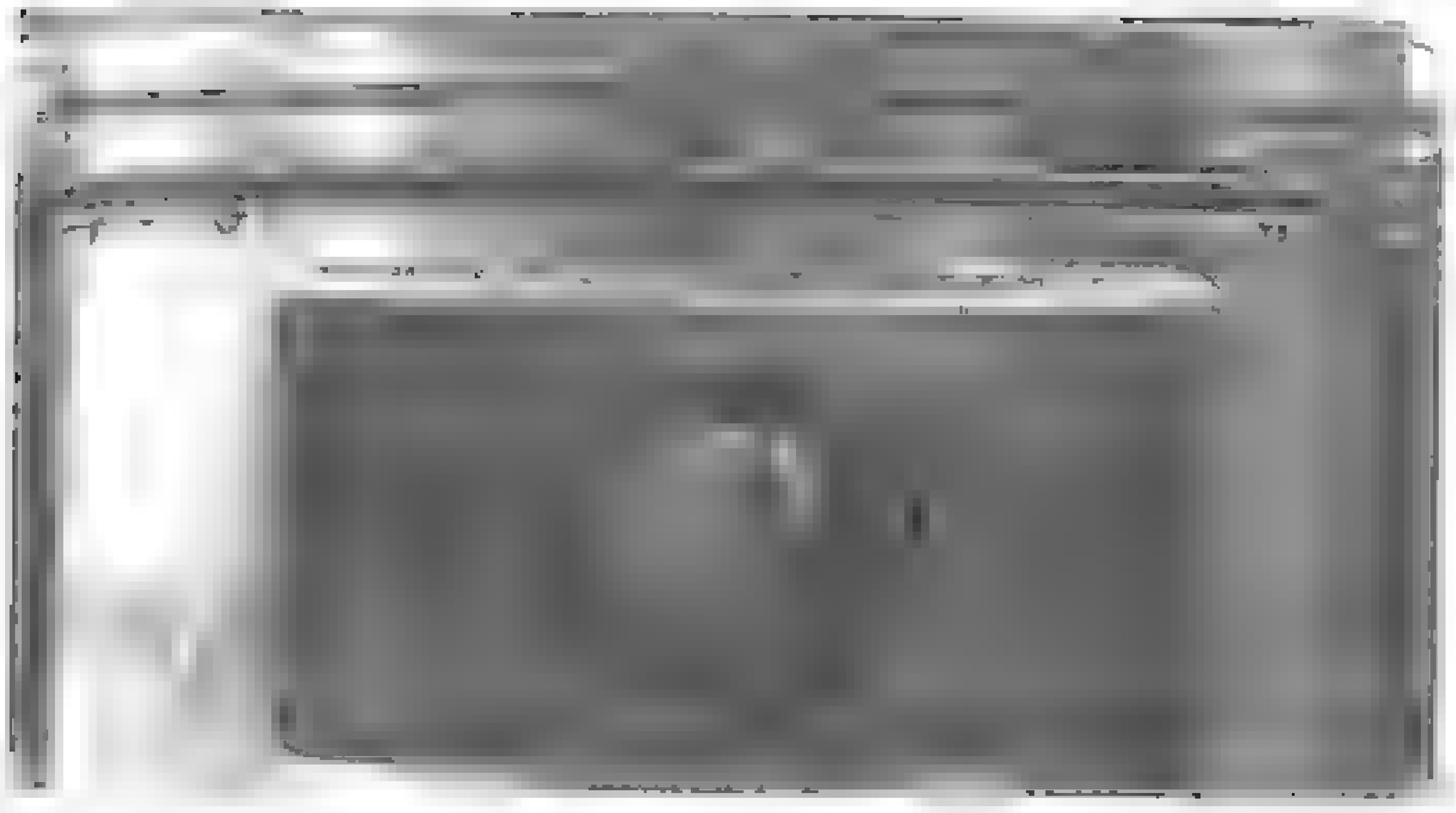
The piston pin is retained by a variety of means in the production engine, the [illegible] common being either an interference [illegible] fit in the connecting rod, or by flat circl[illegible] If the latter are used, they should be coated with Loctite. Ensure that the circlip groove [illegible] clean and undamaged so that each circlip can seat securely. The interference [illegible] (0.0015–0.0025in) is suitable for racing engines, but care must be taken to asse[illegible] and remove the piston from the rod without causi[illegible] any piston damage.

As racing engine speeds are now moving past 8,000rpm, many engine builder[illegible] are experiencing problems with circlips (either flat or round wire). At very high rpm or if the e[illegible]e detonates they may close u[illegible] due to the forc[illegible] erted on them, and drop out of the piston into the bore. To [illegible] it is necessary to employ [illegible] means of pin retention. Some have reverted to interference press fit, but generally I [illegible] to use double Spirolox. [illegible] Spirolox retainers.

The fit of the pin in the piston exercises more control over [illegible] is often appreciated. The ad[illegible] a piston by c[illegible] [illegible]dified if the piston is unabl[illegible] to a tight pin. The pin cl[illegible] [illegible] piston and pin-to-rod) [illegible] [illegible]nance and racing [illegible] to compensate for [illegible] production engine is unas[illegible] experience. At high en[illegible] [illegible] and con rod distortion can severe[illegible] [illegible] the piston pin [illegible] if it has [illegible]ston-to-cylinder scuffing or seizur[illegible] may result. In extreme instances the pin will push the side out of the piston.

For most engines I set a pin-to-rod clearance at 0.0008–0.0015in, and [illegible] piston clearance a little tighter at 0.0005–0.0008in. This will alleviate any bind between rod and pin or between pin and piston that could lead to pisto[illegible]

[illegible] also have the rods checked to ensure that they are not bent, twisted or of[illegible]



Oil drain-back holes that break out below the bottom oil rail improve oil control and reduce parasitic losses. Note the lightweight taper bore gudgeon pin

THE PISTON RINGS

The piston ring, as it slides up and down the cylinder wall, has to function like a bearing but also has to seal off the combustion chamber to keep the gases from escaping past the piston into the crankcase. Most engines use either two or three rings per piston.

The first or top ring is called the compression ring; its purpose is to contain the combustion pressure so that maximum power can be obtained. This ring also has the burden of dissipating most of the piston crown heat (about 80%) to the cylinder wall.

Some racing engines do not use a second compression ring, but if yours does, it has the task of backing up the top ring in sealing off the combustion charge and removing heat from the piston crown. However, its primary function is to support the oil ring in scraping excess oil off the cylinder wall.

The bottom ring is the oil scraper or oil control ring. Its job is to scrape oil from the [illegible]nder wall, but ensure that enough oil remains behind to lubricate the upper rings [illegible] assist in sealing.

As indicated in Table 11.4, the oil control rings cause a good deal of en[illegible] friction, and we all know that friction costs hp. Therefore many manuf[illegible] to supply oil rings in three grades of tension. Standa[illegible]sion rings are those mostly fitted to ordinary road engines by car makers, but low [illegible]nsion oil rings are now bein[illegible] fitted to some production cars and are the [illegible]ed option in competition engines: they usually provide oil control almost as good as standard tension rings, but with reduced parasitic losses. High-tension oil rings are rarely used, but may be necessar[illegible] in some engines running at very high rpm with poor rod-length-to-stroke ratios.

Some engine builders have been dropping the second compression ring in the hope of gaining some reduction in paras[illegible]c losses. This appears to be an easy route t[illegible] increased hp, but the power g[illegible]ns are only small [illegible] [illegible]ly at high rpm. In fact, it [illegible] [illegible]e power due to reduced oil control. W[illegible]n done properly I am find[illegible] power gains in the order of 0.7% at 8,500rpm, and the gains increase at h[illegible]

speeds. Personally I would not run a two-ring piston in an engine that was running at less than 8,500rpm, and then only with crankcase vacuum applied. Frequently, at these lower engine speeds, the largest gains come from being able to run with a shorter, lighter piston. This can then permit a lighter rod, reduced big end width or diameter etc.

RING TENSION AND RING WIDTH

Let's take a closer look to see what goes on when the compression ring seals off the combustion chamber (Figure 11.8). Many feel that it is the ring's inherent radial tension that holds it against the cylinder wall. As the piston descends on the inlet stroke that is correct. If there was very little radial tension the piston would not be capable of pulling a good vacuum; so radial tension is critically important. However after ignition it is primarily gas pressure behind the back of the ring that forces the ring face against the cylinder wall to provide an effective seal.

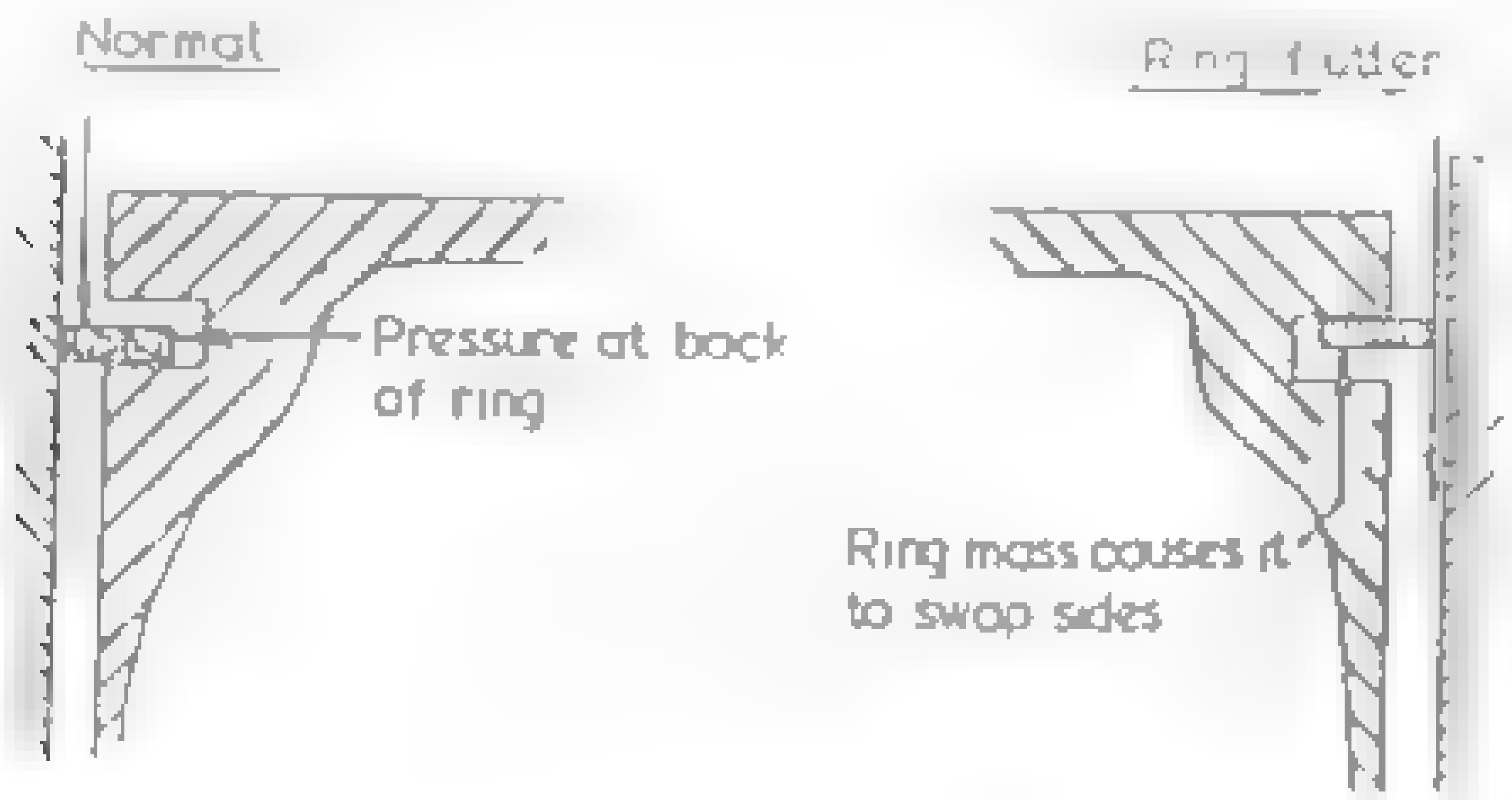
Table 11.4 Comparison of piston ring function and friction

	Pressure sealing	Oil control	Frictional losses
Top ring	78%	5%*	30%
Second ring	22%	45%*	20%
Oil ring		70%	50%

* Top and second rings only assume such high levels of oil control when oil control rings are worn excessively or sticking

A detrimental phenomenon that may occur in a high-rpm engine to reduce the ring's sealing ability is ring float or flutter. As the piston approaches TDC it is slowed down by the connecting rod, but the rings try to keep on moving, and if they have

Figure 11.8 Only the top ring is shown to illustrate how combustion pressure behind the ring forces it against the cylinder wall to effect a good seal. When ring mass causes it to swap sides in the groove, gas pressure is unable to get behind the ring.



enough weight each will break contact with the lower side of its groove and bang into the top of the groove. When this happens, the ring seals off the gas pressure in the combustion chamber so that it cannot get behind the back of the ring and force it against the bore wall. Any gas pressure that may have been behind the ring quickly [illegible] into the crankcase

Radial tension in the ring is unable to prevent blow-by caused by ring flutter. However, a certain degree of radial tension is necessary for good sealing, otherwise [illegible] the ring off the cylinder wall. This would allow blow-by and it is this that we normally see occurring when the rings are old and have lost their tension

Ring flutter allows blow by, but it can also wreck engines. When the ring loses contact with the bore wall it is unable to transfer heat from the piston to the water jacket. The end result can be melted pistons or severe detonation, due to increased combustion temperature.

The narrower a ring for a given radial depth, the higher the rpm at which this effect commences. This has led many tuners to believe that they should use very narrow rings in all competition engines. However, narrow rings have a disadvantage as was mentioned earlier, the compression rings are the primary heat transfer path for the pistons Reducing the ring width will slow heat transfer, so the piston may overheat and soften The top ring land may distort and jam the ring, which will result in the hp falling off, or worse the piston will melt

The other negative aspect of narrow rings is their reduced service life This may not be a concern in sprint engines if you have the time and cash to replace rings regularly. However, in an endurance engine rapid wear of the top ring could see a serious drop off in power before the race concludes, hence my recommendation to keep right away from narrow rings unless they are really necessary

For the majority of high-performance road and competition engines running up to about 8,000rpm I recommend either 1/16in or 1.5mm top and second compression rings Engines running up to 9,500rpm would run a 1.2mm or 0.043in top ring and either a 1/16in or 1.5mm second compression ring. Above 9,500rpm I would usually drop the second ring and retain a 1.2mm top ring, and only at 11,000rpm or higher [illegible] necessary to move it to 0.260–0.300in from the piston crown to reduce the heat load on the ring. The oil ring should then be placed 0.220in below the compression ring

With oil control rings the tendency also has been to go to very narrow three-piece rings and, surprisingly, no attempt is being made to use the space made available to move the gudgeon pin higher, which raises the question of why bother with narrow oil rings? If there is no problem with sufficient space between the piston pin and the top of the piston I prefer to run with a 3/16in or 4mm oil ring pack Such a wide groove allows for four good size oil drain-back holes on both sides of the piston. Along with a full piston skirt with a square bottom edge to scrape oil off the cylinders, such good oil drain-back ensures that the top ring will seal effectively rather than 'aquaplaning' over a deep film of oil A good top ring seal also means more hp, and less oil in the combustion chamber reduces the likelihood of power sapping trace detonation Only in very limited circumstances would I consider going down to a [illegible] or 3mm oil ring [illegible]

then I would insist on large 3.2mm oil drain holes that actually broke out of the bottom of the oil ring groove by around 1.2mm so as to drain oil from below the lower rail.

PISTON GAS PORTS

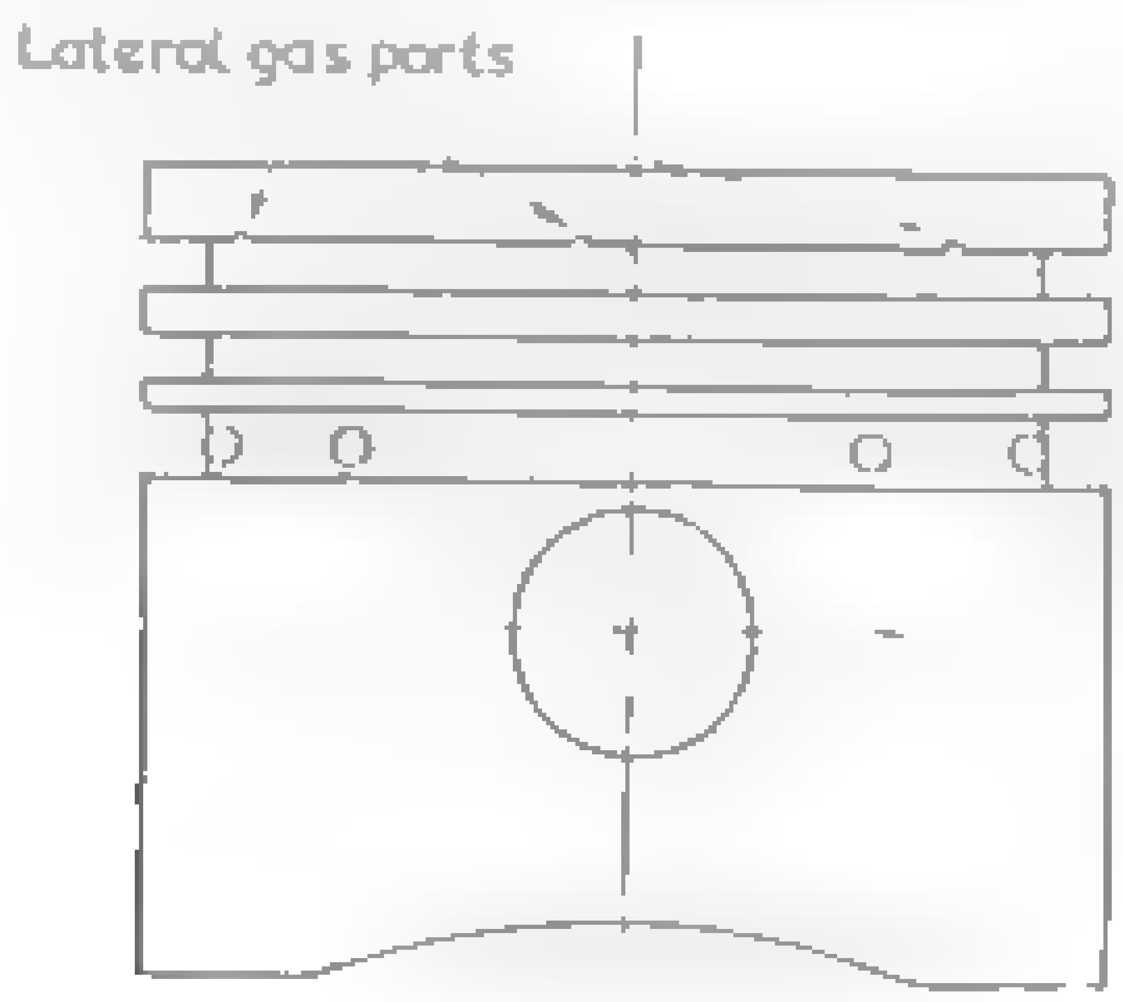
Apart from reducing the weight of the top ring, by reducing its width, there is something else we can do to help keep away from ring flutter in competition engines. By adding a series of lateral gas ports above the top ring we are able to maintain gas pressure behind the ring, even if the ring does swap sides as the piston moves past TDC. In effect these gas ports help an ordinary piston ring work like an 'L' section Dykes ring by ensuring that there is always an open path for gas pressure to get in behind the ring and push it out against the bore wall (Figure 11.9).

In earlier times builders of drag race engines routinely drilled many gas ports from the crown of the piston down through to the back of the ring groove. Unfortunately this practice exposed the piston and ring to a lot more heat, so component life was dramatically shortened. Lateral gas ports are nowhere near as aggressive, even so I recommend that no larger than a 2mm bit be used to mill a half round port to the full depth of the ring groove. The number of these ports will vary according to piston diameter. Below 83mm we should not exceed 6 ports, up to 93mm this would increase to a maximum of 7 and up to 105mm and 115mm we increase to no more than 8 ports and 9 ports respectively.

PISTON RING LIFTING

This brings us to another problem that many tuners do not appear to understand. As well as the top ring being thrown up to the top of the ring groove as the piston rapidly reverses its direction of travel at TDC, it can also be pushed up by pressure building up between the top ring and the second ring. What happens is that gas escaping past the

Figure 11.9 Lateral gas ports allow combustion pressure to get behind a ring that has swapped sides in the groove and force it against the bore wall.



face of the top ring and through the gap between the ends of the top ring may not get past the second ring and into the crankcase as quickly as gas is building up in this area. This pressure will then work in opposition to combustion pressure which is pushing the [illegible] the ring breaks full contact with the bottom face of the ring groove and combustion pressure then escapes under the top ring. That in turn increases the volume and pressure of gas trapped between both rings, further reducing the effectiveness of the seal between the lower surface of the top ring and the bottom face of the ring groove.

Clearly this scenario is going to cost us hp because gas pressure that should be driving the crankshaft is simply leaking into the sump. However that isn't the end of [illegible] its heat to the lower part of the piston. With this part of the piston now much hotter it is unable to act as a heat sink and pull some heat out of the top of the piston. Even more serious is the fact that the top-ring and piston are not making good contact for quite some time and as this is the piston's primary heat rejection path the piston overheats and in serious cases fails. Before that point is reached the engine will be down on power as the hot piston will heat the inlet mixture, and to keep out of detonation the spark advance will have to be reduced.

There are two things we can do to keep gas pressure from lifting the top ring. First up we must ensure that the second ring has sufficient end gap. Everyone frets about losing gas pressure through the ring gaps and into the sump. Therefore we have worked to bring the ring gaps down smaller, but we need to keep in mind that there is a limit to how far we can reasonably go. A race engine with thick, rigid cylinder walls should have a second ring gap at least as wide as the top ring. If the cylinders are prone to flexing a little then you need more gap in the second ring.

The second thing we can do is increase the space between the two rings to provide a larger reservoir for gas to escape into. In a race engine we gain a little extra volume anyway as we use a thicker ring land, but in addition to that we can machine a gas balance groove into the ring land. This groove may appear insignificant but it adds considerable volume to the area and as such reduces any tendency for pressure to build up and lift the top ring.

Never do we want the pressure to equalise above and below the top ring. If that happens we only have the weight of the ring to keep it sealing against the bottom of the ring groove, which isn't anything like sufficient. In fact this is part of the reason why when we add vacuum to the crankcase our horsepower rises. We cover this in more detail in the following chapter, but basically what we find is that when we keep crankcase pressure below zero by means of a scavenge pump drawing blow by gas out, then we gain hp.

Part of this increase is because there is less friction. The low pressure environment means that we don't have a couple of litres of oil clinging to the crankshaft or suspended as droplets in the crankcase area for the crank and rod big-ends to smash into and thus cause drag. Additionally the vacuum literally pulls oil out of the cylinders. Therefore pistons and oil rings draw off less power scraping oil out of the bores. Because the oil rings have less work to do we may be able to drop the ring tension right off and cut parasitic losses even more.

The other portion of the power increase is due to the bigger pressure differential existing above and below the top ring increasing the effectiveness of the seal between

the face of the ring and the cylinder, and also between the bottom of the ring groove and the lower edge of the ring. A better seal means a bigger push on the crankshaft plus it ensures best possible heat transfer out of the piston to the water jacket.

ZERO GAP SECOND RINGS

At this point some readers may be questioning how the above information can be accurate when zero gap second compression rings are held up by some race engine builders as being good for power. Clearly with no gap the second ring's escape route for blow-by gas is quite restricted and this must be causing the top ring to lift. However to my mind the merit of zero gap second rings relates more to how they help those engines operating at extremely rich air fuel ratios achieve some semblance of engine life. These rich mixtures may be necessary to cool the engine internally, or they may be the product of crude injection systems forced on us by race regulations.

When we have an engine running very rich methanol or nitro mixtures cylinder lubrication is marginal. Added to that a huge volume of fuel can escape past the rings into the sump, severely diluting the lubricating oil and exposing the engine to wrecked big-end bearings and cam lobes/tappets in particular. When zero gap second rings are [illegible] the rings. The rings and cylinder bore then bed in much better and so effect a superior seal. Additionally because the lube oil is not diluted to such a degree the engine makes more power after a couple of hours' operation than an engine with conventional rings due to the cam lobes being closer to their new shape and the engine bearings being healthier. As well as keeping parasitic losses lower, bearings in good shape help to subdue crank harmonics and this in turn improves the accuracy of the valve timing events and ignition firing events.

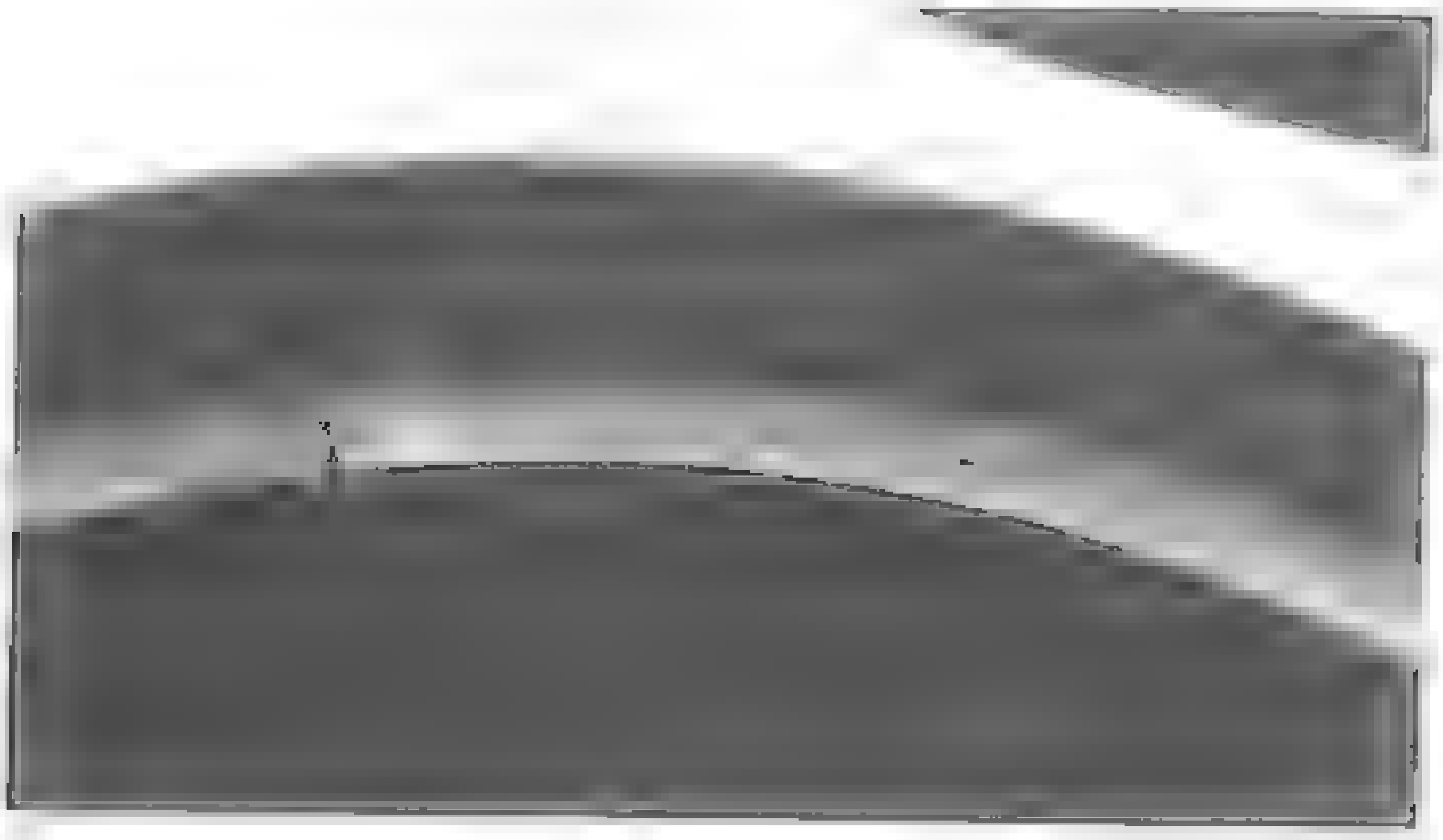
Consequently while it may appear on the surface that zero gap second rings offer many benefits, such is not the case. Outside the areas I have just mentioned, ie when extremely rich alcohol or nitro mixtures are present, they are not recommended.

However do not assume that I am opposed to the idea of gapless top rings. Now that the technology is available to machine the two pieces, called the primary ring and the rail, to more exacting tolerances necessary in a top ring they can work very well in [illegible] gains as high as 5%; I think it is not above 2% at best, but any gain is a help at any time performance is not limited by traction.

RING END GAP

I have always accurately positioned the ring ends so as to reduce gas and oil leakage through the ring gap, even though some maintain this isn't necessary in a race engine. It has been proven that the rate of ring rotation to crank speed is about 1:1000. In other words an engine operating at 7,000rpm will have the compression rings rotating at 7rpm. Hence the idea that it's a waste of time staggering the gaps. However I still maintain that the ends of compression rings should be staggered 180°, and oil rail should have their ends staggered at least 90°.

Leakage through the ring gaps and blow-by (not flutter-induced blow-by) is more of a problem at low rpm, simply because there is more time for the gases to find



Running ring gaps too narrow is not wise in a high hp engine. Typically such engines ... more heat into the rings due to more time spent at wide open throttle; perhaps due also to the use of less co... on piston material and higher placement of the rings on the piston. The 'dimple' near the ring end indicates that this side of the ...

their way through the ring gaps. This is why high rev engines usually require only one compression ring, providing the oil control ring can look after oil control without the assistance of a second compression/s...

This in no way implies that we can simply unw... a new set of rings and st... em straight onto pistons without first checkin... end gaps. Actually the converse is ... e to be extremely careful to fit each ring perfectly square in its bore and ... sure the end gap and increase it if nece...ry by filing the ring ends. Th... ring gap per inch of bore ... follows: air-cooled, turbo and supercharged ... burnin... alcohol ... 0.006in. This is suitable also for l... nitrous doses, but ... competition nitrous ... ngine will want this incre...ed to 0.0065in p... ch of bore. Oil ring rails must b... east 0.004in per inch of bore diameter, but ... larger gap won't hurt. All other eng...

... Engines with hypere... a lot more heat into the top ring ... nce some manufacturers are reco... top ring gap be incre... when these pistons are fitted. I find that this increase is about right for road engines b...

...ine is rebuilt if there is no sign of ring end polishing, indicating the ends a... touching due to insufficient ... p by 0.002in. Then if at the next ... uild there is still no indication of the ends touching reduce the ... ap another 0.002 ... ple will no doubt question such large ring gaps but let me assu... that when the engine is actually running at full power the top ring gap will close up ... 150 ... e due to the heat put into it. If the engine gets hot enough to detonate, ... l go close to closing right up. One way you can check if you have suff... ... by inspecting your old rings. If the ring ends are polished the end ...

touching, therefore increase the gap by 0.002in. If you don't you could break the rings, severely score the bores or ruin the pistons!

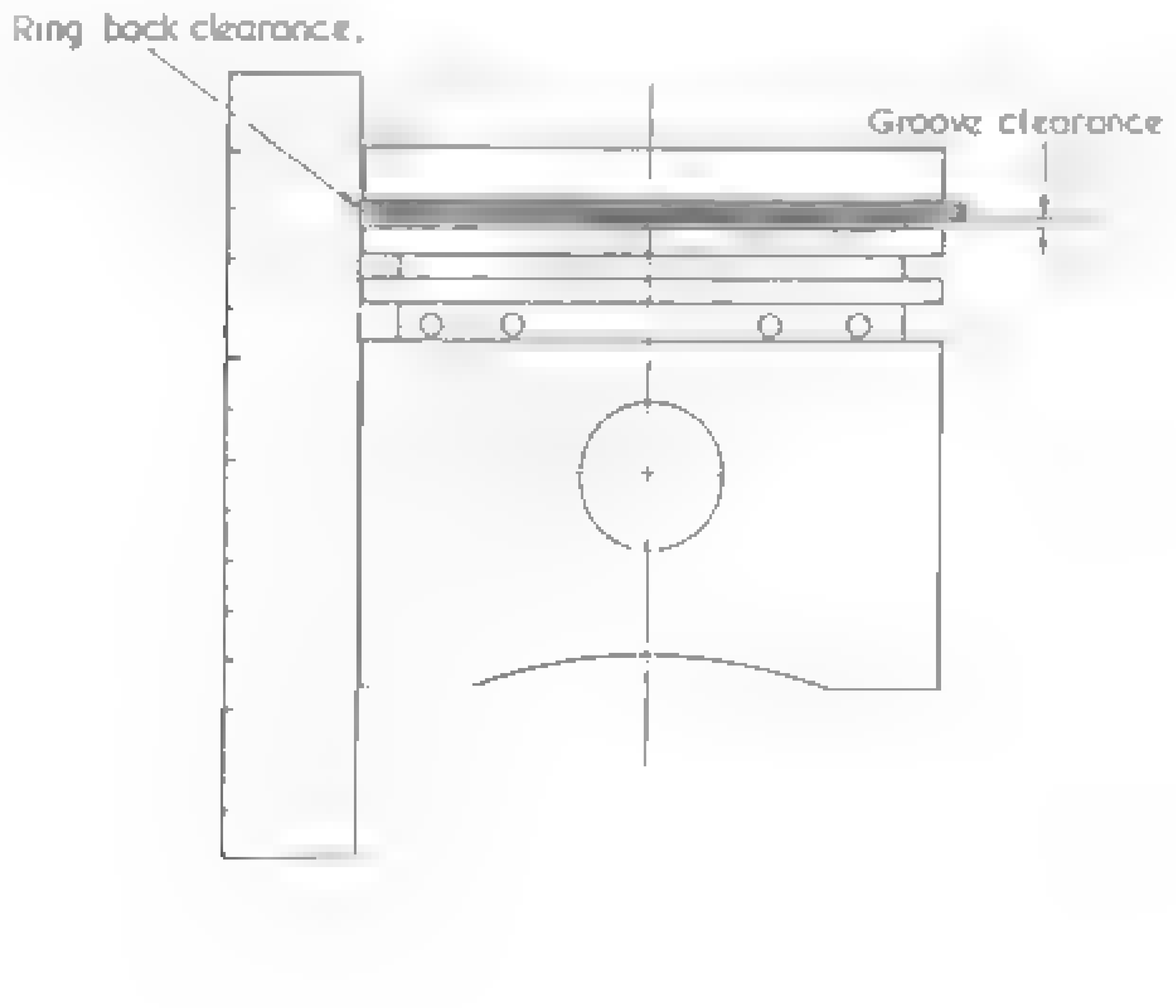
In practice most blow-by is due to bore distortion and leakage past the ring faces rather than through the gaps, so be careful when it comes to setting the end gaps. Also keep in mind that once sonic velocity is achieved through the ring gap the flow volume will not rise with an increase in combustion pressure or supercharger boost. Therefore as we tune our engine to higher performance levels blow-by through the ring gaps will not increase. Overall blow-by though will worsen, but this is due to more bore distortion and increased leakage past the ring faces.

Some people will point to superior leak down test figures as evidence that zero gap second rings and smaller gaps, especially on the second rings, is the route to less blow-by and thus more hp. Let me assure you that in spite of what you may have previously been led to believe, there is virtually no relationship between extremely low leak down numbers and increased hp.

GROOVE CLEARANCE

These days ring breakage is rare and can usually be attributed to worn piston ring grooves that allow the ring to flop around and break, or excessive taper of the bore, causing radial flutter and subsequent ring breakage. If new rings are fitted to a worn bore, and a ridge is present at the top of the bore, the new rings will bang into the ridge

Figure 11.10 Before rings are fitted the back clearance and groove clearance should be checked. With used pistons carbon may build up in the groove, causing ring jamming and breakage. Excessively worn grooves allow the rings to twist and flop about, unable to seal effectively and likely to break.



and break. Excessive piston-to-bore clearance can lead to ring failure due to the piston rocking and twisting the rings as it passes TDC.

Generally, a racing engine should be set up with very tight groove clearances of 0.001–0.0015in. Certainly there should not be more than 0.002in side clearance between the ring and ring groove. A road motor should ideally have the same groove clearance as a racing unit, but up to 0.0035in clearance is acceptable. If we have chosen a top-of-the-line premium race engine piston/ring package, we can go very tight on groove clearance. An unrestricted engine should be held at 0.001in ± 0.0002in, while a naturally aspirated inlet restricted engine can usually tighten up to 0.00065in ± 0.00015in, unless your supplier advises otherwise for your particular application.

Also check for ring back clearance as shown in Figure 11.10. This is vital if you are re-using a piston as you will very quickly find if a ring land has closed up and is jamming the ring anywhere. And what can appear to be just a light layer of carbon in the back of the groove may in fact be deeper than it appears and jam a new full depth ring hard against the bore wall.

RING DESIGN AND MATERIAL

In the past compression ring material was usually plain grey cast iron, either plain, chrome-plated or moly-filled. However, most top rings are now ductile nodular cast iron, which is much more durable and heat resistant, and less prone to detonation cracking than plain grey iron. There is nothing wrong, though, with using plain cast iron in the second ring position.

Generally for the top ring I prefer Sealed Power/Speed Pro moly-filled nodular cast iron rings. Nodular iron has almost three times the strength of conventional grey cast iron. It is ductile rather than brittle and can be bent without breaking. Molybdenum belongs to the same chemical family as chrome; however, it has a lower coefficient of friction and higher resistance to abrasion. Its thermal conductivity is several times greater than that of either cast iron or chrome plated cast iron, and its porosity acts as an oil reservoir, reducing scuffing and cylinder wear.

For oil control I like the multi-piece pressure-back type. The idea is to keep frictional losses to the very minimum while at the same time maintaining adequate oil control. Therefore only low tension type oil control rings should be used for race engines. The Sealed Power/Speed Pro low-tension oil ring is excellent in this type of application. However if I'm using a top spec piston, usually from Mahle or JE, I will order a complete piston and ring package. Mahle, for example, source their rings not only from their own manufacturing facilities, but also from other European and Japanese manufacturers, so as to provide the best product for a particular application.

HOW TO FIT PISTON RINGS

When rings are being fitted, care is needed to avoid fitting them the wrong way up and to prevent damage by incorrect installation. Compression rings can be permanently twisted if they are fitted in the groove at one end and gradually screwed around until the entire ring is in place. Instead, they should be expanded sufficiently to fit over the piston, then allowed to drop into the groove. Special expander tools are available for

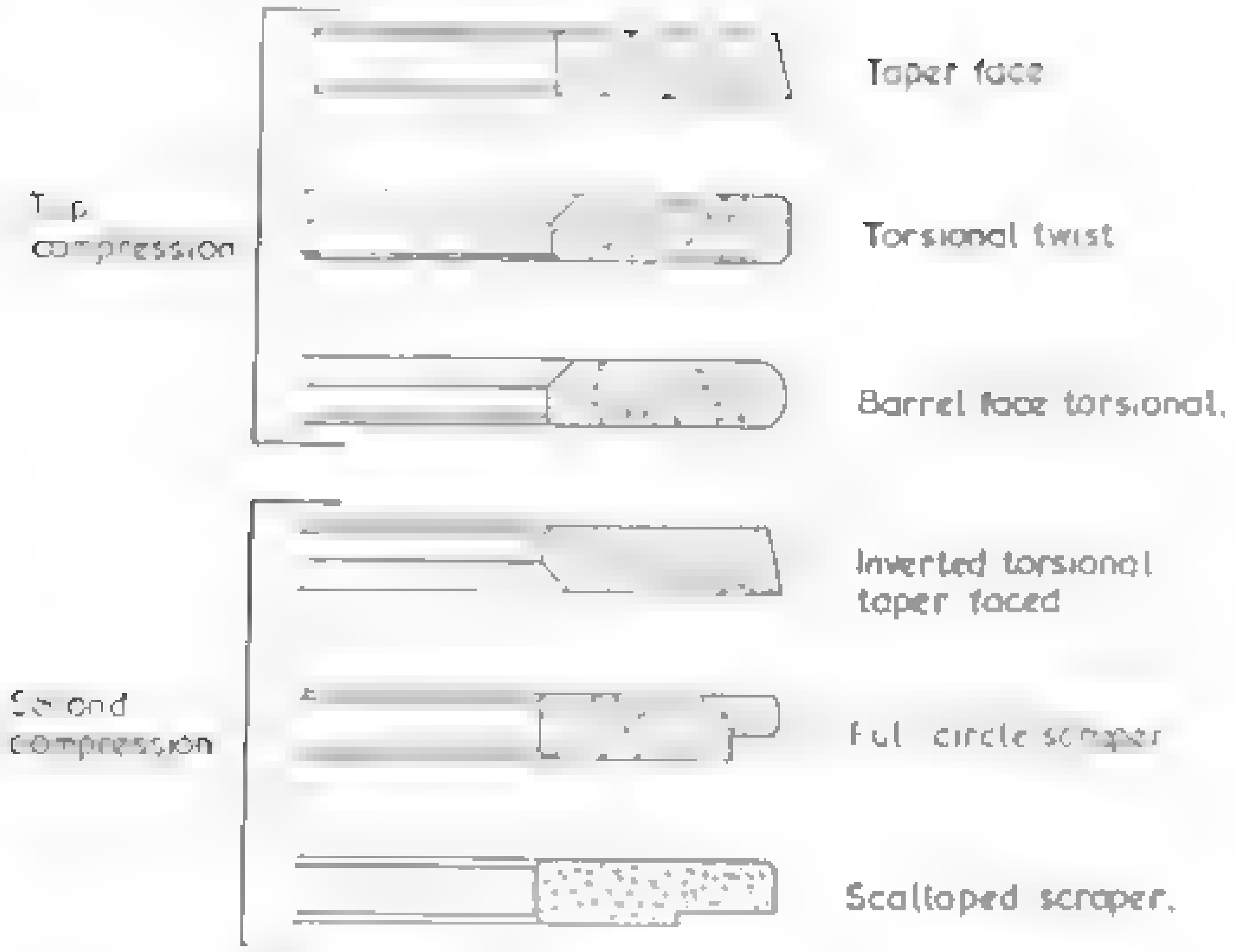


Figure 11 11 Rings must be fitted the correct way up to effect a good seal and maintain proper oil control

[illegible] of the ring (Figure 11 11)

The unbalanced section or torsional twist ring is fitted with the inner chamfered edge uppermost. This causes a slight dish in the ring face and provides the same characteristics as the taper-face ring, in that the lower edge makes a high pressure contact with the cylinder wall. Whereas the taper-face ring eventually wears parallel, the torsional twist ring retains its characteristics

The 'Multo Seal' type second compression ring is fitted with the serrations in the ring face downwards

All multi piece oil control rings should be fitted in accordance with the packaged instructions. Always double check that the ends of the expander ring have not been lapped over

Before fitting the assembled pistons in the engine, dip each entire piston in engine oil and oil the walls of each cylinder Some feel that the oil should be fortified with some additive like STP, some even paint straight STP on to the cylinder walls and pistons. This is wrong Piston rings are not designed to cut through an additive like this, the end result could be glazed rings and bores

BEDDING-IN RINGS

The initial bed-in of rings is achieved by giving the car a full throttle burst for a few seconds, followed by snapping the throttle shut and coasting for a few more seconds. This should be repeated at least 12–15 times with the engine at normal operating temperature. Accelerate the vehicle in top gear from the slowest speed it will pull in that gear.

By giving the engine full power, the high gas pressures force the rings out against the bore wall. Snapping the throttle shut causes a vacuum in the cylinder, which draws up extra oil. This and a low engine speed minimises the risk of glazing and allows the ring face and cylinder wall to cool.

After the rings are initially bedded-in, the engine can be operated at up to 80% of its rev/power potential, but constantly vary the speed. If this is not done, the rings may still glaze. After about half an hour of this on the dyno or the race circuit, the rings can be considered as being run-in. A road engine with chrome rings should be run in for around 200 miles, preferably in one session. Again, constantly vary the engine speed and include heavy acceleration for short bursts as for initial bedding in. Avoid constant high speed until the engine has done 500–700 miles.

I prefer to break in my engines on the dyno, carefully controlling the engine speed and load to ensure long engine life and maximum power. This takes 1–1½ hours for most types of engine. When the engine cools, the tappets are adjusted and the head is re-tensioned. Then, following a warm-up to bring the water and oil up to temperature, full load power tests are made to determine ignition advance, fuel jet sizes, etc. By the end of the tests the engine is fully run-in (Table 11.5).

Table 11.5 Standard break-in procedure on an engine dyno

rpm	Engine load (torque lbf ft)	Time (mins)
3,500	40	10
4,000	50	5
4,500	70	10
5,000	85	15
5,500	100	20

Note. this is the procedure used for all race and rally Ford BDA engines of 1,600–2,000cc producing maximum torque of 135–165lbf ft. Other engines require a different run-in load according to their displacement and torque output.

Some feel that it is preferable to run an engine on the dyno from some external power source. Theoretically this should smooth out the differences in fit and surface finish, but this solves only part of the problem. Internal temperatures influence the fit and shape of parts, and to further complicate matters clearances will alter with mechanical loads as well as with temperatures. It follows then that the engine must be run-in under its own power, initially unloaded, then progressively loaded to produce the required running surface.

TESTING RING SEAL

When full power testing is being carried out, a blow-by gauge should be connected to the crankcase, which will give a quick overall indication of how well the rings are sealing.

With the amount of scavenge I usually run on the dry-sump pump I expect to see the blow-by gauge reading less than 1–1.5cfm. If the power is down and the gauge is reading higher than this, I can be pretty certain that there is a ring seal problem somewhere.

An alternative is to block off all the engine breathers except one, then connect a 'T' into the open breather hose. Connect one branch of the 'T' to a 'U'-tube water [illegible] of water in a carefully built engine with excellent ring seal, then on another occasion it is very easy to quickly check on the ring seal. If, for example, the engine is down on power and the water gauge is showing a blow-by pressure of 9in with the same size breather restrictor as previously used, you know that the ring seal is not good.

Obviously, for this latter method to be of any value, a leak-down test must be done first. Then, having established that all cylinders have an average leak-down of, [illegible] initial manometer 'calibration test' to establish what size of breather restrictor is required to produce a blow-by pressure of 2in of water will have real relevance. These [illegible] pumps. Consequently running a sealed crankcase I expect to see negative crankcase pressure, even at the end of endurance engine tests.

LEAK-DOWN TEST LIMITATIONS

The leak-down test is a much less accurate indicator of ring seal, so I prefer to use it [illegible] cannot accurately reflect what the ring seal is actually like when the engine is operating; it can only point to where a problem may exist. Note that this test will also [illegible] in which area of the engine the problem lies.

The actual leak-down test is fairly straightforward. After bringing the engine up to temperature, pull all the spark plugs, then bring No 1 piston up to TDC on the compression stroke, connect the leak-down tester in the spark plug hole and connect the compressor air line to the tester. The leak-down tester will then display a percentage air loss reading. Ideally we want only 2–3% leakage, but 4% is acceptable. [illegible] up to TDC and tested in turn for leakage.

Really, any serious competitor should do a leak test at the race track as soon as a practice session is concluded, and likewise at the end of competition for the weekend. If the leak-down shows up a problem, look for it and fix it before your next race. Some [illegible] because it so badly needs attention. What they should have done was to save their money for meals, accommodation, fuel, etc, and spent it to give the engine a light hone and a set of rings, and touched up the valve seats and perhaps replaced the valve springs and bearings.

HARMONIC BALANCER

The pulley or harmonic balancer attached to the nose of the crankshaft has its part to play in the durability of an engine, and the surrounding environment. Many four cylinder engines use a pressed metal drive pulley, spot welded or riveted to a hub. At higher engine speeds, ie 6,000rpm-plus, a pulley of this construction is liable to fly apart, wrecking anything in its path. To avoid this, a cast or machined component should be used.

The harmonic balancer is fitted to dampen crankshaft torsional vibrations that could, if not controlled, wreck the crankshaft. At times the balancer and pulley may be in one piece, or a separate pulley may be bolted to the harmonic balancer. Again, if this pulley is a pressed metal component, scrap it to avoid problems.

The actual harmonic balancer is made of three parts bonded together. A rubber belt is bonded between the outer inertia ring and the hub. Failure is normally caused by the inertia ring losing its bond with the rubber. If this occurs at high engine speed you can be assured of spectacular and expensive results. To prevent this, the harmonic dampener must be carefully inspected on a regular basis. If you race, do it after each race meeting. To assist in your inspection you should mark a distinct, common line between the hub and inertia ring. If the two marks move out of line, it indicates that the ring is moving on the hub, so scrap it. Under no circumstances should you try to repair the balancer by bolting the inertia ring to the hub. This will wreck your crankshaft just as surely as running 2,000rpm over the limit. The inertia ring must be able to move, within limits, to dampen crankshaft vibration.

There is little that can be done to improve the reliability of the harmonic balancer except to check it for run-out before fitting. The other modification necessary, if a fully counterbalanced crank is fitted, is to machine away the counterweight cast as part of the hub.

Before you fit the harmonic balancer (or pulley) to the engine, paint it black. After fitting, accurately find TDC and file a good groove into it, to align with the TDC mark on the engine timing case. Paint the groove with silver or white paint. If you also paint black the area around the timing marks on the engine timing case, and the TDC and ignition timing marks or pointer silver, the ignition timing will be much easier to adjust later on. To adjust the timing accurately you must be able to see the timing marks easily, so anything that you can do to make them stand out must help.

FLYWHEEL

The flywheel is attached to the other end of the crankshaft. To ensure that it remains attached, two good dowels (3/8in) and retaining bolts that have been coated with Loctite and correctly torqued provide good insurance. To cut costs, manufacturers seldom fit dowels, so it will be necessary to machine the flywheel and crank to fit these.

Over the years there has been a fair amount of controversy with regard to flywheel lightening. Some claimed that a lightweight flywheel improved acceleration due to reduced inertia, others stated that the only benefit was better crankshaft life because the twisting load on the end of the crank was reduced, and there was less risk of flywheel explosion due to the inertia load on the centre of the flywheel being lessened. At this time we can state and prove that any reductions in an engine's

reciprocating mass or rotating mass will improve acceleration. Thus lightweight pistons, rods, cranks, clutches and flywheels all benefit performance

The problem previously was that we had no reliable means of measuring the performance advantages of light components. However, the situation changed when relatively inexpensive computerised engine dynos like the Superflow SF-800 became available These dynos can accelerate an engine at a fixed rate and accurately record the horsepower. This means that we can programme the dyno to accelerate the engine at, say, 2,000rpm per second, and record the power levels every 250rpm. When such a test was carried out on a 388cu in Chev speedway engine, it recorded almost 25hp more when titanium rods and a light flywheel were fitted The driver claimed that the car was quicker by about a half length out of turns, which backed up what the dyno had showed. Tested with a static load as we had to with the older dynos, the engine showed no difference in hp when the lightweight parts were fitted

A very light flywheel is undesirable for a road vehicle as it can produce a lumpy engine idle. The reason manufacturers use a relatively heavy flywheel in the first place is to absorb the uneven torsional impulses coming through the crankshaft and keep the engine turning smoothly at low engine revs. A fairly hot engine with a relatively heavy flywheel is much more pleasant to drive on the road than one with an overly light one

If you decide to use a lighter flywheel it is not always necessary to buy a new lightweight version or even have your old flywheel turned down. Some manufacturers have, over the years, built many engines of various capacities around the same basic design Therefore the flywheel fitted in production to the 1,000cc version may also fit the 1,500cc derivative and save possibly 5–7lb, if you have the 1,500cc engine in mind for modification, you can use the lighter flywheel from the smaller engine

Lightening should only be done near the outside of the flywheel. Removing metal from the centre is fatal and likely to lead to failure. When metal is removed from the outside it makes life easier on the centre of the flywheel as the inertia loads are reduced. This is important as some flywheels are none too strong. To prevent distortion from heat, do not machine away so much of the clutch side of the flywheel that there is only a thin ribbon of metal left, just wide enough to provide a seat for the clutch plate friction material. If the clutch is subjected to severe usage, that thin ribbon may distort, due to localised heating, and allow clutch slip

A light flywheel improves acceleration on the track but can make street driving unpleasant

Chapter 12

Lubrication

In the past the lubrication system was simply that – a means of pumping lubricating oil between vital engine components to prevent metal to metal contact, and premature wear or seizure. Today's engines, however, demand much more of the lubrication system as the lubricating fluid is called upon to perform a bigger role in cooling the bearings, pistons and valve springs. This is especially true of high output engines as there are clear limits on how much heat we can move out of the piston crown to the water jacket via the piston rings. The only alternative to this route is to utilise the lubricating oil, literally squirting it up into the top of the piston to transfer heat from the piston to the oil.

However, in performing the dual roles of lubricating and cooling, the oil ends up in large quantities in places where we don't want it. The reason we don't want it in certain places is because this increases parasitic losses – costs us horsepower and impedes engine acceleration. Additionally we don't want it in certain areas because while there the oil is whipped up, becoming aerated and thus unable to carry out its lubricating and cooling activities so effectively on its next pass through the engine. Consequently when we think of engine lubrication we have to extend our focus well beyond just the lubricating oil in isolation. We have to look at the entire delivery system, plus the oil return system and oil cooling system. Also we want to examine ways of enabling the oil to do its work more efficiently while winning back hp previously lost to parasitic losses.

Obviously stock or lightly modified engines will work quite well without any modification of the lubricating system. And as manufacturers take parasitic losses ever more seriously in their quest for higher hp and reduced fuel consumption, the opportunity for us to make gains in this area in road cars is becoming smaller. Therefore for some readers their prime concern will be the lubricating oil itself. They will be desirous of choosing an oil that minimises engine wear and provides a power gain by cutting friction and other parasitic losses.

Apart from its lubricating and cooling roles the engine oil has a number of other basic tasks that we most probably don't even think about. These are: provide a seal

between piston rings and cylinder, permit easy engine starting from cold, keep the engine clean of sludge, minimise combustion chamber deposits, not 'poison' the lambda probe and catalytic converter, provide rust and corrosion protection, and resist foaming. To indicate just how well an oil performs in these areas, various test organisations around the world assign engine oil for street use a series of letters and or numbers to tell us what tests the oil has passed. While there is a question mark over the validity of such tests, they do give the average consumer some idea of how well a particular oil does its job.

OIL TEST STANDARDS

For example the American Petroleum Institute (API) assigns petrol engine oils a service rating such as SD, SE, SF etc. Currently the toughest API rating is SM, and to carry that rating an oil must pass a series of tests with the engine running on an engine dyno. One test monitors the oil's ability to prevent valve train wear in an ohc 2.3-litre American Ford. Like its 2-litre European cousin this engine is notorious for valve train problems, in its early life being subject to a recall for abnormal cam lobe wear. Zinc is a great lifter/lobe anti-scuff agent, but unfortunately as it does nasty things to lambda probes and CATs it had to be removed from the oil additive package. An equivalent substitute has yet to be found, so all street oils now struggle with this test. (Race oil and some motorcycle oil formulations are superior in this regard. They don't have to meet car manufacturer's demands regarding lambda sensor and CAT life, so zinc can still be added by the oil companies.) Another test monitors the oil's capacity to prevent wear and resist oxidation in a 350cu in Oldsmobile run for 64 hours with the oil at 149°C sump temperature. Poor quality oils finish up as tar by the end of this test, but the better oils, usually full synthetics, will still look like oil at the finish of the 64 hours. In fact some oil companies now duplicate this test and extend it up to 128 hours to show how well their synthetic oils hold up. Consequently while the majority of oils available may achieve an SM rating, literally just scraping through to gain a pass, others will easily pass and a few will far exceed the test standard.

The oil packaging usually lists some of the test standards that the oil has passed.

It is these latter oils in which we should be interested. Each will have its strengths and weaknesses. Others may not be outstanding in any particular area, but may be close to the top in a number of categories. Generally it would be just such an oil that I would choose. However for a special situation I may also select an oil with that particular attribute which I deemed to be absolutely essential.

VISCOSITY AND HORSEPOWER

For example, my usual oil of choice for road engines is Mobil 1, 5W-50 grade viscosity. It provides a very high level of engine protection first and foremost. There are one or two other oils about which are similar, or even marginally superior in this regard, but they are not so readily available and they are considerably more expensive. Also these are down on power and show increased fuel consumption. The horsepower king, while still providing adequate engine wear protection is the 0W-40 grade of Mobil 1. As you can see in Table 12.1 when tested against a large number of oils, including some with lower viscosities, it was a clear winner. In comparison with Mobil 1 5W-50 it has slight fuel economy benefits too, in the order of 1–1.5% under highway cruise conditions. Therefore when it is maximum hp that I am after 0W-40 tri-synthetic is the grade I choose. However it allows slightly more engine wear than 5W-50 and if the power advantage is to be maintained it has to be changed more frequently as it thickens up more rapidly due to a significantly higher evaporation rate.

Table 12.1 Synthetic oil horsepower homparison

Rpm	Mobil 1 5W-50	Shell Ultra 5W-40	Castrol R 10W-60	Amsoil 10W-40	Mobil 1 0W-40	Royal Purple*	BP Visco 5W-40	Castrol SLX 0W-30
3000	143.2	144.8	140.5	141.2	142.9	140.7	139.8	138.6
4000	202.9	201.3	203.6	203.2	202.4	202.5	202.3	201.2
5000	263.1	262.5	262.4	262.5	266.6	262.8	261.5	260.8
5700	290.4	290.0	293.3	290.7	297.4	294.0	289.9	289.7
6200	304.3	303.5	306.5	305.6	311.2	308.7	303.6	305.1
7000	280.7	278.9	282.7	281.4	288.1	284.5	279.1	282.4

* The Royal Purple used in this test did not have a viscosity rating specified, but lab testing shows it to be in the 0W-20 to 0W-30 range.

Note. Test engine was a slightly modified Subaru Liberty RS turbo. Oil and water temperature were controlled at 95–97°C and 85–87°C respectively.

help an engine make more hp. However, viscosity is only half of the picture – friction modifiers are the other half. You would think that a light oil would always guarantee a hp benefit, but as you can see when you compare the two Castrol products that isn't the case. The thicker 10W-60 Castrol R synthetic made more maximum power than most of the oils tested, while its thinner sibling was close to the bottom in overall hp. Clearly if we are looking to gain hp from a particular type of oil then there is more to it than simply choosing one with a very low viscosity.

UNDERSTANDING VISCOSITY AND VISCOSITY IMPROVERS

engine turning over

pumping resistance of a thick oil will give a substantial decrease in power

of 50 SAE at high temperatures.

Large advances have been made in polymer chemistry, but these VI improvers

BLENDED AND SYNTHETIC OILS

However, adding polymers or seeking out special mineral oil base stocks is not the

stock with mineral oil base stock to produce a blended synthetic. Such an oil is [illegible] to produce a high-grade, wide-viscosity oil suitable for use in both high-performance [illegible]

[illegible]

Synthetic oils offer many benefits in competition engines over even the very best mineral and vegetable oils. Because they are so thin when cold they lubricate better [illegible]

[illegible]

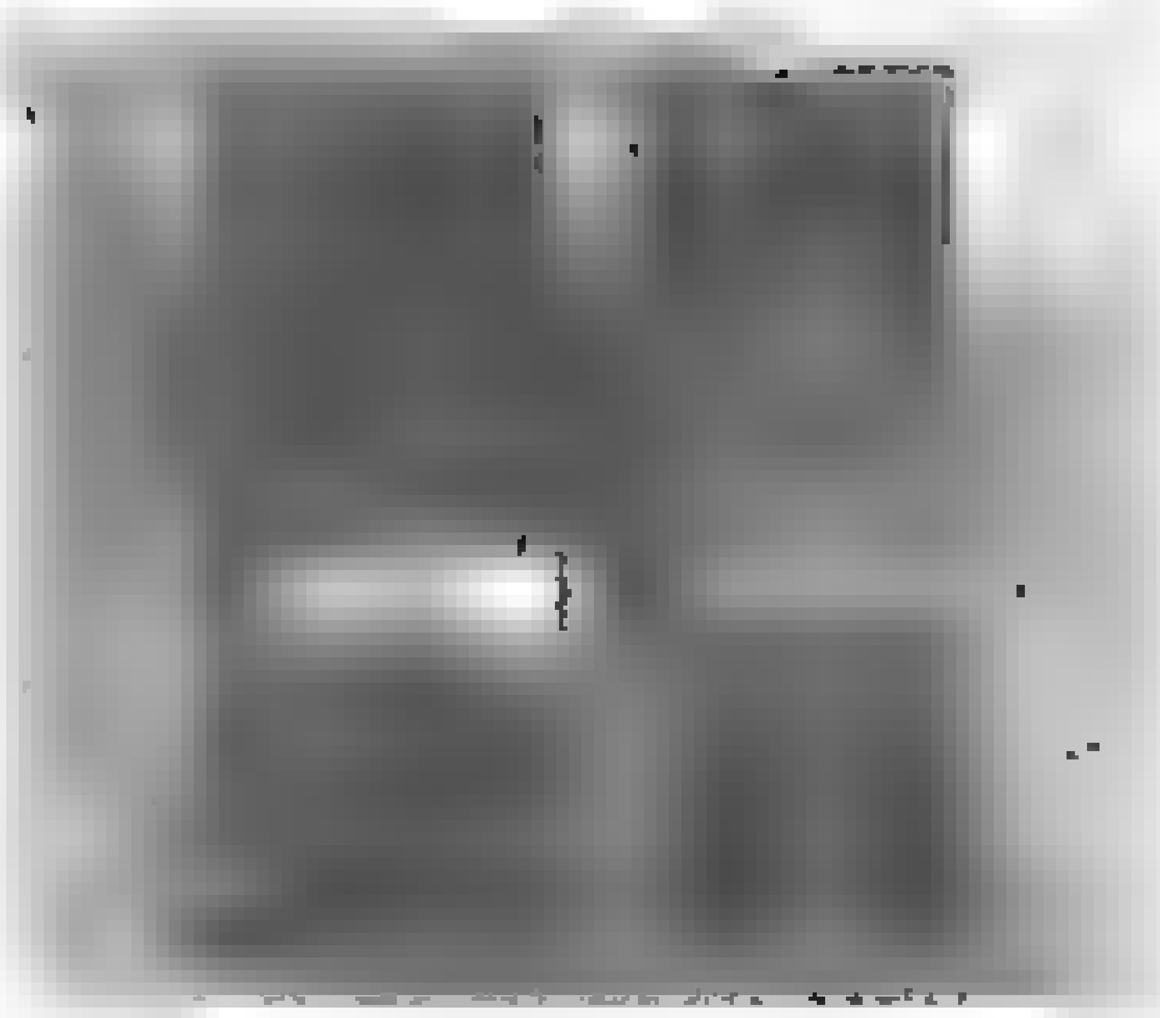
maximum output competition engines, but for road engines and less developed race [illegible] Mobil 1.

[illegible]

OIL ADDITIVE PACKAGE

Earlier it was mentioned that a considerable part of engine oil is in fact an additive package. Beside polymers, oils also contain a number of other compounds. Some work as anti-oxidants, some as detergents, some as foam suppressants, and others are anti-wear additives that in total may constitute 20% of the oil.

These identical upper rod bearings have done the same competition distance in identical engines. The shell on the left, lubricated with competition mineral oil, has more than twice as much wear as the bearing on the right, which came from an engine lubricated with Mobil 1 synthetic



Oxidation occurs due to the presence of oxygen and combustion by-products in the oil and due to the temperature of the oil itself. As the oil temperature increases past 100°C, the rate of oxidation accelerates and the anti-oxidant becomes less effective.

The by-product of oxidation and combustion is known as sludge. To keep this in suspension, rather than allow a build-up inside the engine, detergent is added. Racing oils, however, may contain very little detergent, and for this reason they should only be

Mobil 1 synthetic 15W-50 is preferred for all maximum output competition engines, while 5W-50 (or 0W-40 tri-synthetic) is recommended for all other engines. In sprint where oil does not get time to warm up and stays below 85–90°C use 0W-40 … power loss

used in road vehicles when they carry an API rating such as SJ, SL, SM etc. or the European ACEA rating of A3 or A5

To counter aeration, a foam suppressant is added, which will prevent the formation of foam under the conditions for which the oil is designed to be used, but it cannot eliminate the foam resulting from the presence of water in the oil or from air sucked into the lubrication system either through a fractured pick-up or due to oil surge that has allowed the oil pump to pump air.

Obviously this additive package eventually wears out, and when the anti-wear compounds are depleted engine wear accelerates rapidly; when the anti-oxidants and detergents break down, sludge begins to build up in the engine, and so forth. This situation should be avoided, and it can be if the oil is replaced while these various compounds are still working effectively.

HOW OFTEN THE OIL SHOULD BE CHANGED

How often the oil should be changed will largely depend on the air temperature and driving conditions. Stop-start driving and cold weather under 10°C/50°F are hardest on oil, while long distances between stops with little or no engine idling in warm weather (20°C/70°F) is very easy on today's oils. In the first instance I would be changing Mobil 1 every 4,000 miles or four months, whichever came first. In the latter situation Mobil 1 would provide good protection up to 12,000 miles or six months however, I would change the filter at the midway poi

In race engines how often you change the oil will probably depend on your budget more than anything else. However there is no good reason to throw away synthetic oil after every race meeting. Obviously if you have just done a 24-hour event you would toss the oil, but for most engines there is probably no need to replace the oil any more frequently than when the engine is stripped for a rebuild. Really when you get your oil tested for the presence of metal and contaminants the report should state how the viscosity has changed and if the additive package has been depleted

One major concern with oil in competition engines is fuel contamination Rich [illegible] heated at 90–100°C to boil off the fuel In fact some barbecues see more use cooking fuel and water out of oil than cooking meat!

OIL TEMPLRATURE AND OIL COOLERS

No matter how good your oil is, it must be maintained at the correct temperature to lubricate effectively. A good deal of engine wear takes place because engines are operated with cold oil and/or water. An engine should never be driven hard until the oil reaches 70°C, while the ideal operating temperature is 95–110°C. It can go as high as 130°C for short periods, but oil breakdown and excessive oxidation will take place above this temperature. In a race engine, bearing failure is a possibility any time the oil temperature goes past 130°, and generally hp is lost over 115°.

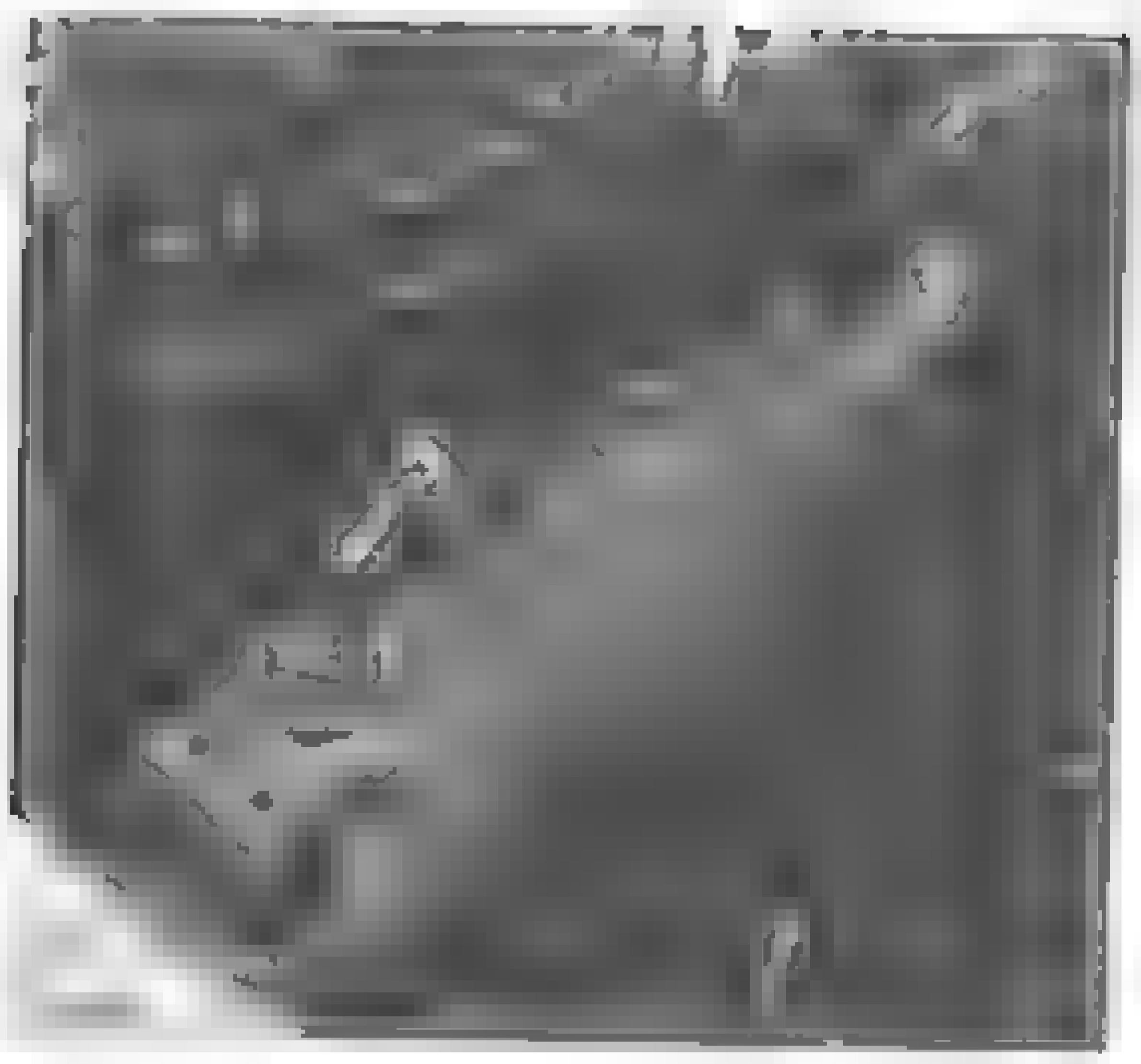
Years ago an early modification was to fit an oil cooler, but with advances in oil technology this is often no longer necessary for street engines, and also sprint type competition engines An exception would be when we are limited in what we can do to [illegible]

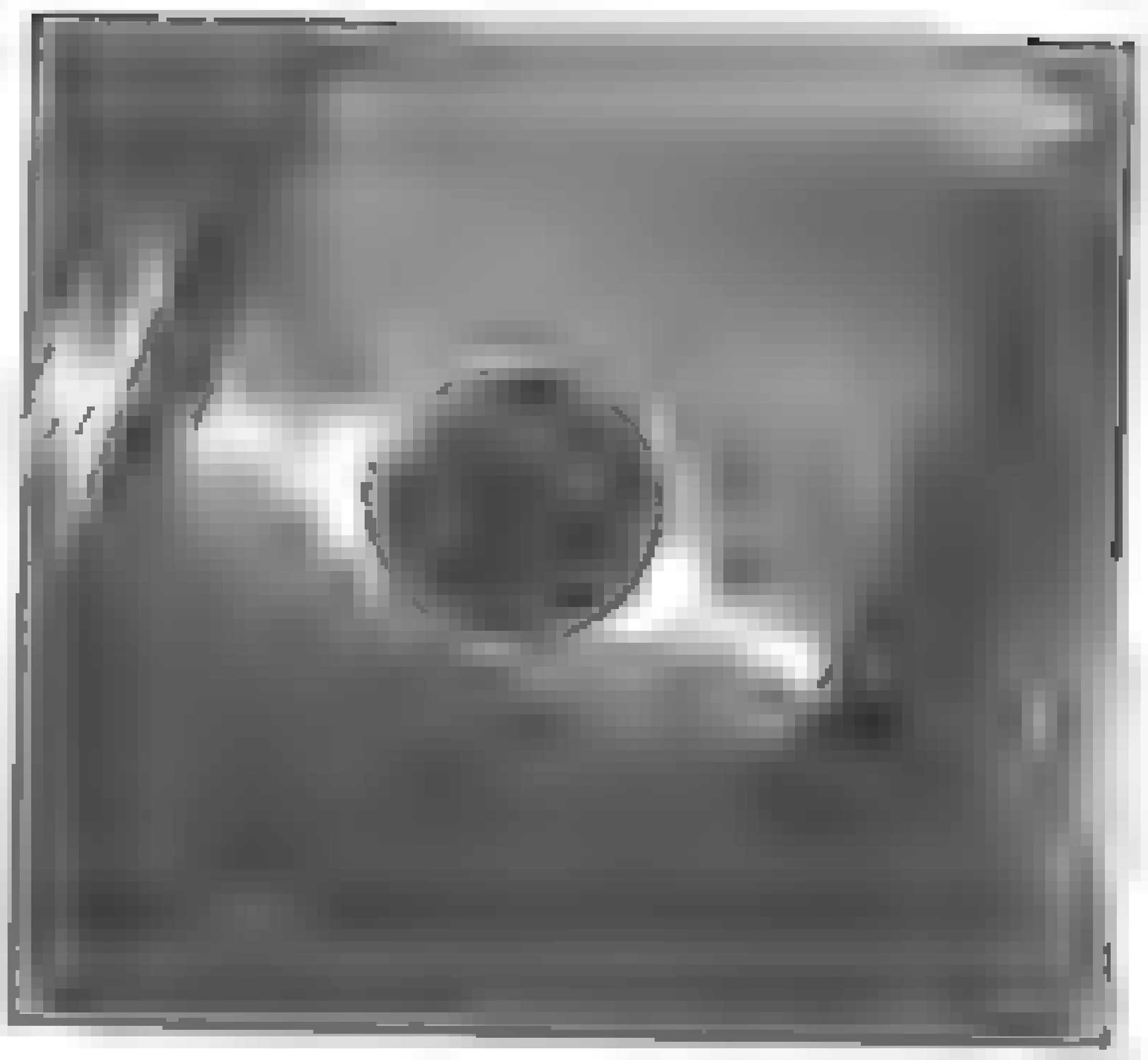
bring water cooling temperatures down without a large outlay of money and/or major re-engineering of the plumbing system. In such situations the simple addition of an oil cooler could help to get enough heat out of the engine that no change to the cooling system is needed.

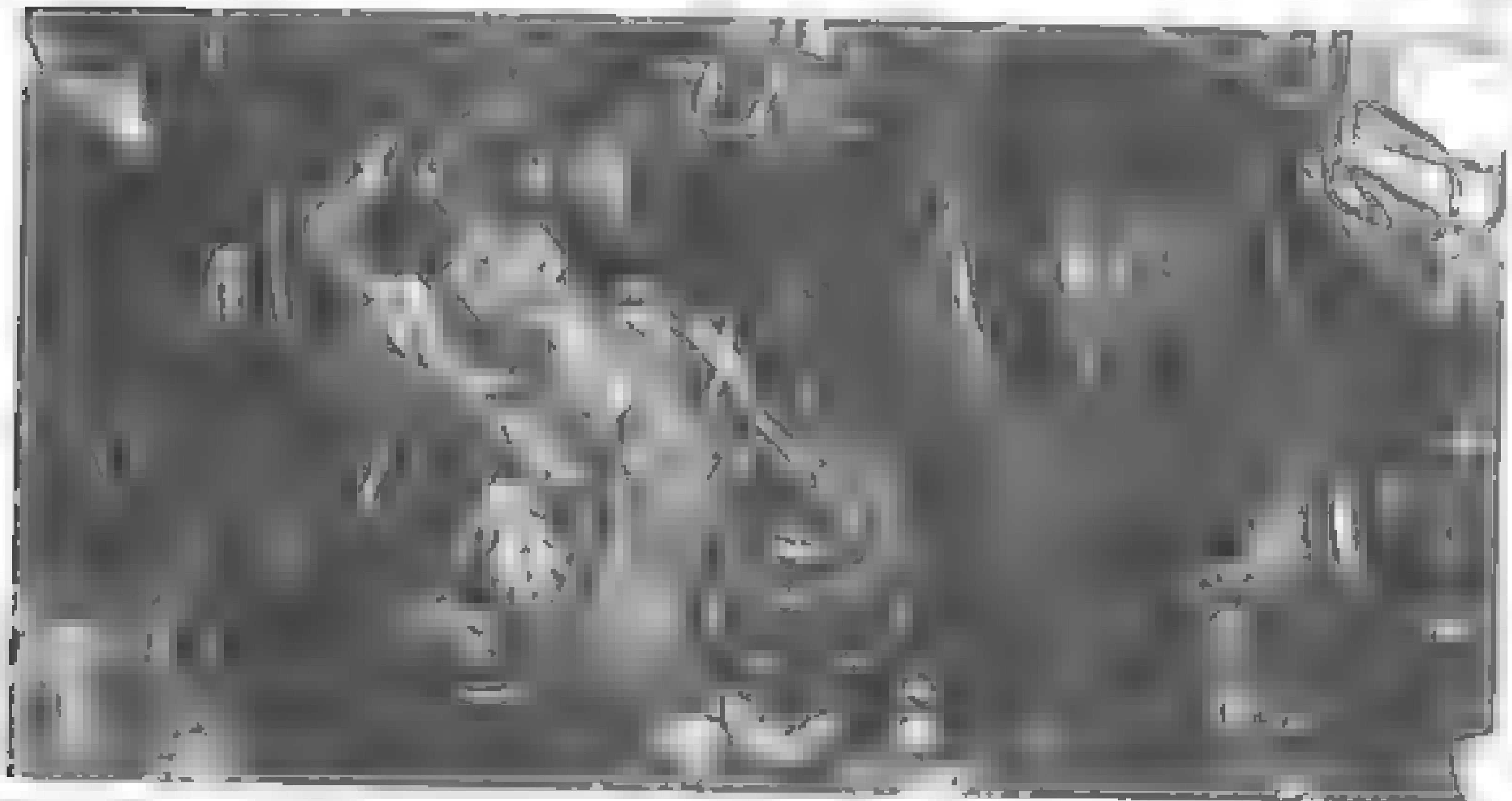
At the completion of big events you at times see a podium finisher express his amazement in standing there due to a problem in the oil cooling system sending the oil temperature sky high for much of the race. After thanking the sponsors, which invariably includes an oil company, it is often difficult to determine if the tale of the high oil temperature was a free advertisement on national TV for the sponsor, or if it was for real. However, I know of a race car completing several hundred miles of an endurance event with the oil cooler bypassed after accident damage. Temperature in the oil tank went as high as 170°C but on strip down engine components did not appear to be in any worse condition than if temperatures had been normal. That was on full synthetic 15W-50 Mobil 1. However I have also seen full throttle 24-hour track endurance tests by manufacturers with the same mineral oil in their sump as their new vehicles in the showroom. During such tests sump oil temperatures of 150°C and higher, up to 180°C, for the entire test have been recorded. Yet subsequent engine teardown revealed that such high temperatures were not a problem. This is not an endorsement of very high oil temperatures, but they do show that at least for short periods a new engine, or one still with very good piston ring seal, and within spec bearing clearances, can operate at 120–130°C sump oil temperatures. I would only recommend an oil cooler if under hard driving conditions the oil is regularly in excess of 120°C. I base that recommendation on the fact that I have seen a number of highly stressed race engines lose bearings when the oil went past 130°C.

When an oil cooler is necessary there are several 'musts' which should be kept in mind. As already noted, cold oil wrecks engines and deteriorates quickly so in road vehicles either an oil thermostat will be required, or else the oil cooler should have a shutter or blind fitted which can be adjusted to suit driving conditions and the ambient temperature. The oil lines connecting the cooler into the system should be of at least $^1/_2$in bore, preferably $^5/_8$–$^3/_4$in, so as not to restrict flow. Additionally, if a take-off sandwich block fitting between the oil filter pad and oil filter is used, it must be of a non-restrictive design. Many such adapters are simply drilled with passages intersecting at 90°, severely restricting oil flow. The adapters you should select will have oil passages which are cast or CNC machined into the alloy block. They should be finished to look as though they were 'ported' to maximise oil flow. Oil coolers are impossible to clean out so do not purchase a used oil cooler. It could be full of dirt or bearing material. Also if your engine or turbo suffers bearing failure it is always prudent to replace the oil cooler, thus avoiding possible damage from pumping rubbish into your rebuilt engine and turbo.

After fitting the oil cooler, continue to monitor the engine oil temperature. While high oil temperatures may lead to inadequate lubrication, and subsequent failure of bearings, tappets and cam lobes, oil cooled below about 85°C is no good either. In a competition engine vast amounts of money are expended to pick up a few additional hp, but many fail to realise just how much power cold oil is costing them. I stumbled on this when I first began dyno testing my engines. I would shut the engine down to make some adjustment and maybe 15 or 20 minutes later do a dyno pull, and when I did all the calculations (there were no computer print-outs then – we had to calculate







Top left: *If sump oil temperatures regularly exce[illegible] °C, an oil cooler will help engin[illegible]*

Top right: *T[illegible] ...ded up worn like th[illegible]*

[illegible] blocks must b[illegible] ... d are restrictive oil line [illegible] ... es with 90° bends b[illegible]

[illegible] nually) power would be down [illegible] ere. I would quickly do another [illegible] e power would change. What was [illegible] n? During th[illegible] eat, so when the en[illegible] e was tired [illegible] ne th[illegible] ... a number of dyno pu[illegible] on to double-check [illegible] the oil temperature would [illegible] the hp would [illegible] up. Basically I want 5W-50 or 15W-50 grad[illegible] 75–105°C; dropping it down to 85°[illegible]

In hotter climates, high performance road vehicles may also need an oil cooler. It is always a worry recommending an oil cooler for a road car or bike because for a good deal of the time the oil will be over-cooled, particularly during winter. Therefore before you decide on an oil cooler for a road machine, be sure that you really need one. Fit an oil temperature gauge and determine your oil operating temperature in midsummer. If under normal driving conditions your oil temperature is in excess of 140°C, fit an oil cooler. However, if it stays around 110–120°C when you run at full throttle for quite a few miles, you will be wasting your time and causing yourself unnecessary trouble by fitting an oil cooler.

OIL FILTRATION

The next requirement of an efficient lubrication system is adequately filtered oil. Any solid material in the oil will act as an abrasive, wearing away at bearings, crank journals, tappets, cam lobes, etc. The oil filter must remove the majority of this type of material from the oil. Very few lubrication systems are unfiltered these days, but if yours is one of them remember that long engine life is at least partly dependent on clean oil, so steps must be taken to install a filtering unit.

The term 'Micron Rating' is often misused in filtration technology. The important figure is the one that indicates the smallest particle totally removed by the filter. For example, a filter may have to stop all particles over 40 microns from passing through but 5% of particles of 20–30 microns size are allowed through. A system suitable for a racing engine should remove all particles, but this ideal is not possible at present.

A micron is one millionth of a metre – a human hair is about 50 microns thick. [illegible]er the particles circulating in an engine's oil, the more likelihood there will be [illegible]ive wear. The challenge facing filter manufacturers is to stop all of these abrasive particles with a filter that has an acceptable service life. You can do your part by changing the filter regularly, as specified by the manufacturer, and by re-installing

If possible choose a larger capacity oil filter. Both these filters have identical thread and gasket sizes, but one is much larger. This cuts flow restriction and provides more filtration area. If there is insufficient space in the stock location, consider installing a remote mount.

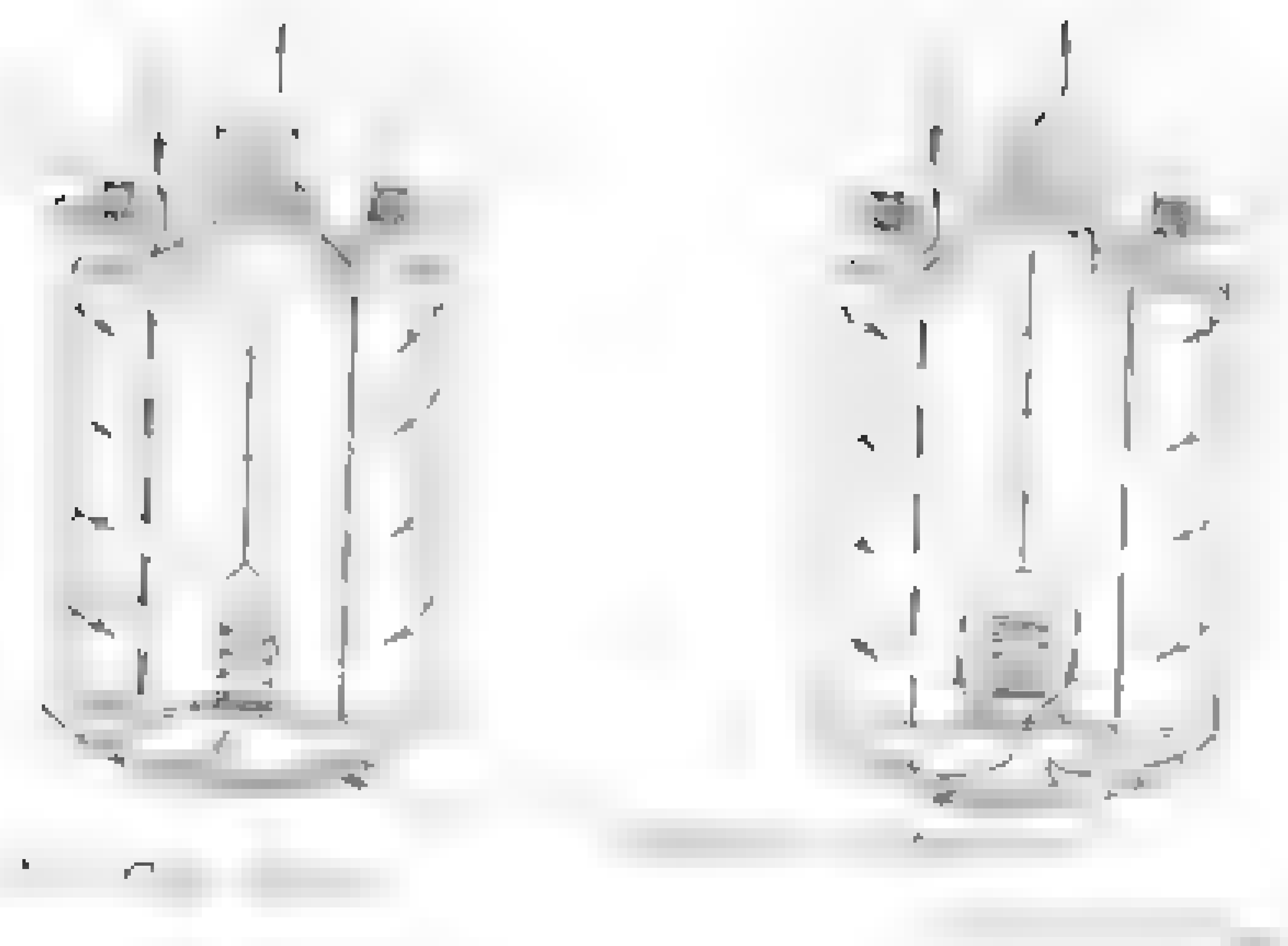


Figure 12.1 Bypass type spin on oil filter

the new filter correctly. If the type fitted to your engine has a replaceable element, be sure to remove all traces of sludge from inside the canister before fitting a new element.

A race or rally engine with thousands of dollars invested in it requires a higher level of filtration than that offered by over-the-counter spin-on or replaceable-element filters. The standard type of oil filter may have a bypass in case it becomes clogged with sludge and other material; this is done for the benefit of the motorist who does not bother to have his filter changed regularly, and also to prevent the filter element from rupturing when the oil is cold and viscous. Once the bypass opens, due to excessive pressure, unfiltered oil is allowed to circulate through the engine. It can [illegible] the engine (Figure 12.1).

A racing oil filter must not contain any bypass and consequently must be burst [illegible] from circulating with the oil without reducing oil flow to a dangerous level, particularly during cold starting.

Many engine tuners cut open their oil filters to judge engine condition, however, I believe in a better system – oil analysis. For a small fee, specialised laboratories and oil companies will analyse oil for the presence of wear metals (indicating filtration effectiveness and/or engine wear), for dilution and sludge (indicating excessive blow-by, incorrect oil type or incorrect oil change period), and other potential problems.

OIL PUMP AND OIL PRESSURE

As engine tune is increased and bearing clearances become greater, the need to investigate the flow potential of an engine's oiling system grows in importance. In lower stages of tune the standard oil pump, if in good condition (check the manufacturer's specifications for gear clearances, etc) will usually be acceptable. It may be possible to increase the oil pressure and oil flow a small amount by fitting a

stronger relief valve spring or by fitting a spacer under the existing spring. A common modification with the four-cylinder British Fords is to increase oil pressure to 45lb with a spacer, or to 60lb by using a stronger spring.

Once the engine reaches semi race tune, a higher-capacity oil pump, which utilises wider gears, will become necessary to keep sufficient oil passing through the engine. At times it may be necessary to change the oil pump drive due to the standard drive not being able to cope with the additional oil pump load.

Some engines suffer oil starvation because of restrictive oil passages. The German Capri V6 and Buick V6 are both good examples of this type of problem. The Capri has an oil passage in the block about 40% smaller than the oil pump outlet. In the very early stages of tune this passage must be enlarged to match the pump outlet if bearing failure is to be avo

In the V8 Chev the restriction in the main oil gallery is the result of a fundamental flaw in engine block machining. General Motors decided that it was simpler/less expensive to bore the passage from opposite ends of the block! Tunnels are put through mountains using this method all the time, with surveyors and engineers keeping the boring operations on course. The tip of an unguided long drill wanders off course, so when boring from both ends of the block the gallery area can be down to 50% where the two holes meet. This is usually not a problem in stock engines but as the power rises so the need to drill the main passage all the way with a very long drill increases.

The V6 Buick has a passage for oil pick up running from the middle of the block to the oil pump located at the front of the engine. This passage is about 25% too small to flow enough oil for a racing engine. After the oil leaves the oil pump, it encounters another restriction, a 90° bend. Following this, the oil is fed into the right lifter gallery, which supplies the right bank lifters, the cam bearings and finally the mains and big ends. Additionally, the front cam journal bleeds off oil for the left lifter bank. All of this bleed-off to the valve gear, combined with the oilway restrictions, reduces the oil flow to the crankshaft bearings. An early modification in the development of this engine must include enlarging the pick-up gallery and eliminating the 90° bend after the oil pump. This engine is a good candidate for an external oil plumbing system that supplies oil directly to the main bearings.

Many engines, in fact, bleed too much oil for the lubrication of the cam and the lifters. To limit the bleed-off, restrictor plugs should be fitted. If you have switched from hydraulic to solid lifters, oil flow to these parts should be restricted by reaming the lifter bores and fitting a lifter sleeve kit.

Some tuners, in an endeavour to supply oil in sufficient quantities to the main bearings, increase oil pressure to 100–120psi. This should never be necessary if the lubrication system is properly modified. In fact, if an 8,000rpm race engine is experiencing any failures at 60–70psi, it usually indicates that work is required to open up oilways and/or restrict bleed-off to the cam and lifters. For example, the Cosworth DFV Formula 1 engine started life running a minimum oil pressure of 85psi. About ten years later after a major rework of passage sizes and angles the maximum pressure required was down to 60psi, and the engine was still reliable at 45psi. This in spite of the engine now running 1,500rpm faster and producing 17% more power. Current Formula 1 engines, producing 920hp and running to 19,000rpm operate at only 30psi. A stock block based competition engine operating above 8,500rpm may benefit if the oil pressure is raised to 80–85psi. The power required to pump oil through an engine is

considerable, so do not increase the power drain to ancillary equipment by unnecessarily increasing the oil pressure

PREVENTING OIL SURGE

bearings. If allowed to go unchecked, bearing failure is imminent

include the thickness of the sump gasket

In engines subjected to the full range of G forces – left and right turns and

right side

Figure 12.2 Wet sump anti-surge baffle

This oil pressure sender switch operates at 30psi and is useful when assessing sump baffles and sump oil fill levels

switch to connect with a 5 watt lamp in the light, otherwise you could be momentarily blinded if the 21W lamp flashes on in the dark

DRY SUMP SYSTEM

The dry sump system avoids many of the pitfalls of the wet sump, so is the preferred lubrication system in competition engines. Figure 12.3 shows a typical dry sump system. The oil is scavenged from the engine and is piped to an external oil tank usually passing through the oil cooler en route. The oil pump will have at least two scavenge stages drawing oil, blow-by and also, if the engine is unsealed, air back to the oil tank. The oil tank is designed to not only store the oil, but also to de-aerate it effectively, for if aerated oil were pumped back into the engine, bearing failure could result. The pressure stage of the pump then draws this de-aerated oil from the oil tank and pumps it through the oil filter and into the engine

The beneficial aspects of this type of system are obvious. The absence of oil surge helps to increase engine life and reduces the risk of bearing failure. The vehicle can be lower without the risk of sump damage, as a shallow oil pan without a reservoir is used. Oil leakage from the engine is reduced as the scavenge stages reduce pressure caused by blow-by, within the crankcase. Depending on the scavenge pumps used crankcase pressure should be less than a couple of inches of water at maximum engine speed and load in a race engine with good piston ring/cylinder seal. This reduced crankcase pressure contributes to a power rise of the order of 3–4%, in spite of the extra hp required to drive a big dry sump pump

The disadvantages of the dry sump system mainly revolve around the expense and additional weight, but also a dry sump system can give many problems if not correctly engineered. The main areas where I see deficiencies are in the design and mounting of the oil tank, and also the sizing and routing of the pressure and scavenge lines.

Figure 12.3 Dry sump oil system with two scavenge stages and one pressure stage. The components illustrated are, 1 scavenge pressure oil pump, 2 dry sump oil pan; 3 oil filter; 4 oil tank; 5 breather catch tank, 6 oil cooler; 7 aircraft screen filter; 8 scavenge lines with inlet screen filters; 9 tank return lines; 10 suction line; 11 pressure line; 12 breather lines

A dry sump system would have prevented the destruction of this engine which occurred on a long high-speed turn. The ... ing could not control ... oil flow interrupted, a rod bearing seize...

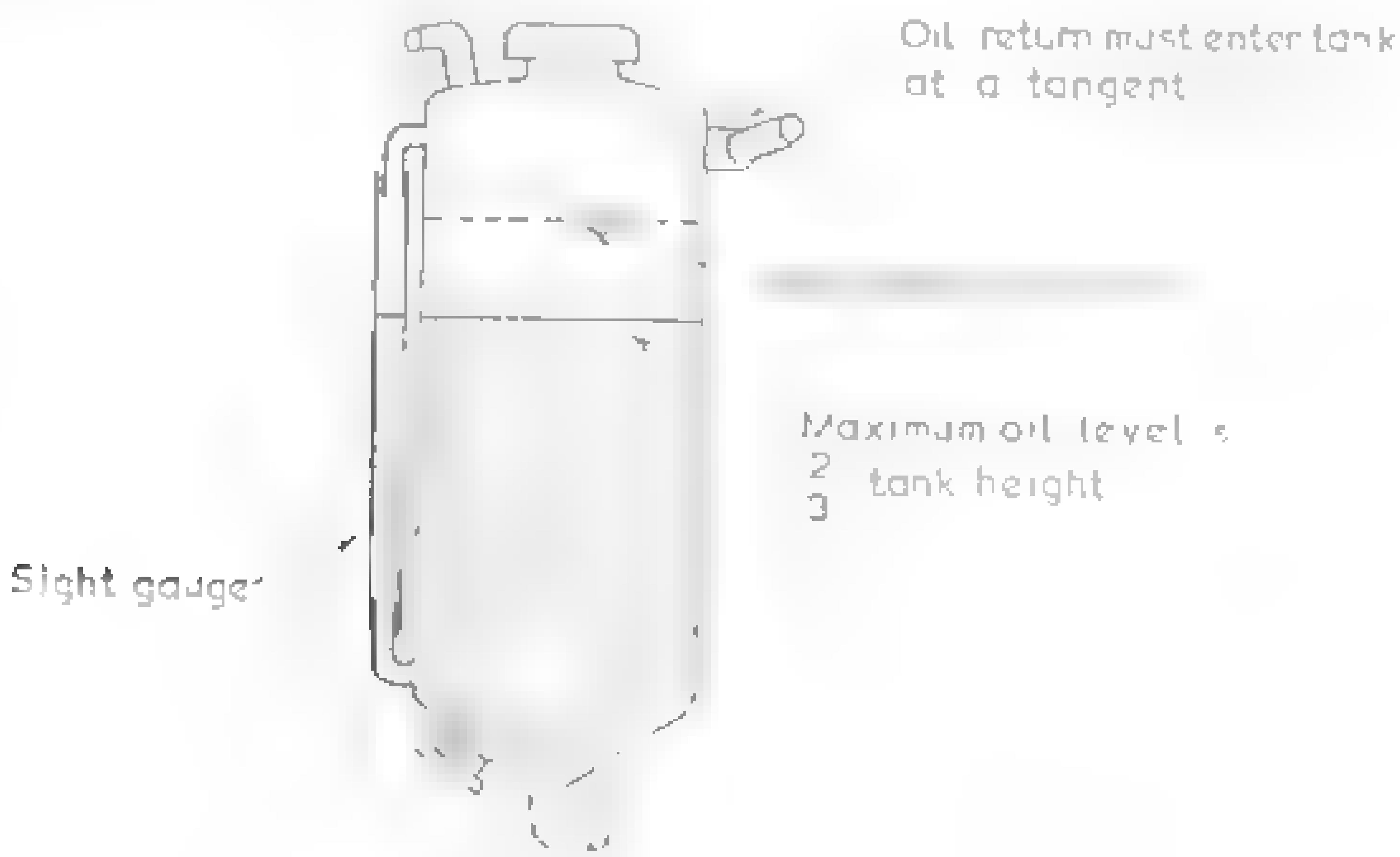
OIL TANK

The size of the oil tank will to some extent be dependent on the size of the engine and the quantity of oil being pumped into it, the effectiveness of the scraper/scavenge system in rapidly returning oil to the tank, the efficiency of the tank in quickly de-aerating the stored oil, and engine oil consumption and race distance.

The oil tank must not be filled more than two-thirds full. With that in mind a properly designed system used in sprint-type competition will use a tank of around 5½–11 litres, with smaller engines running a reservoir up to around 7 litres, V6s up to about 9 litres and V8s up to 11 litres. In distance events consideration must be given to engine oil consumption, not only in the early stages when the engine is 'fresh' and consuming, say, ½ litre/hr, but also in the later stages when consumption may triple. Also think about how long the car will be out on the circuit between pit stops, how time-consuming it will be to top up the oil, and if the regulations limit how much oil may be added. When these factors are weighed up it may be concluded that the sprint tank is suitable, or it could be that up to 30% additional capacity is required.

A tall round oil tank with the oil line entering the top of the tank at a tangent works best (Figure 12.4). With the oil entering at a tangent it will tend to swirl around the tank wall, which assists the air to separate out. To further aid de-aeration a horizontal baffle perforated with 4–5mm holes should be located about midway between the top of the oil and the oil inlet. Then, by keeping the height-to-diameter ratio of the tank greater than 2:1 – preferably around 2.5:1 – the oil will give up more air, reducing the air content from as much as 7% down to around 2–3%. Also, by keeping the tank tall, and with the pump suction line drawing from the bottom of the tank's cone-shape floor, the pump will always be drawing oil into the engine regardless of the G forces being encountered.

Figure 12.4 Dry sump oil tank



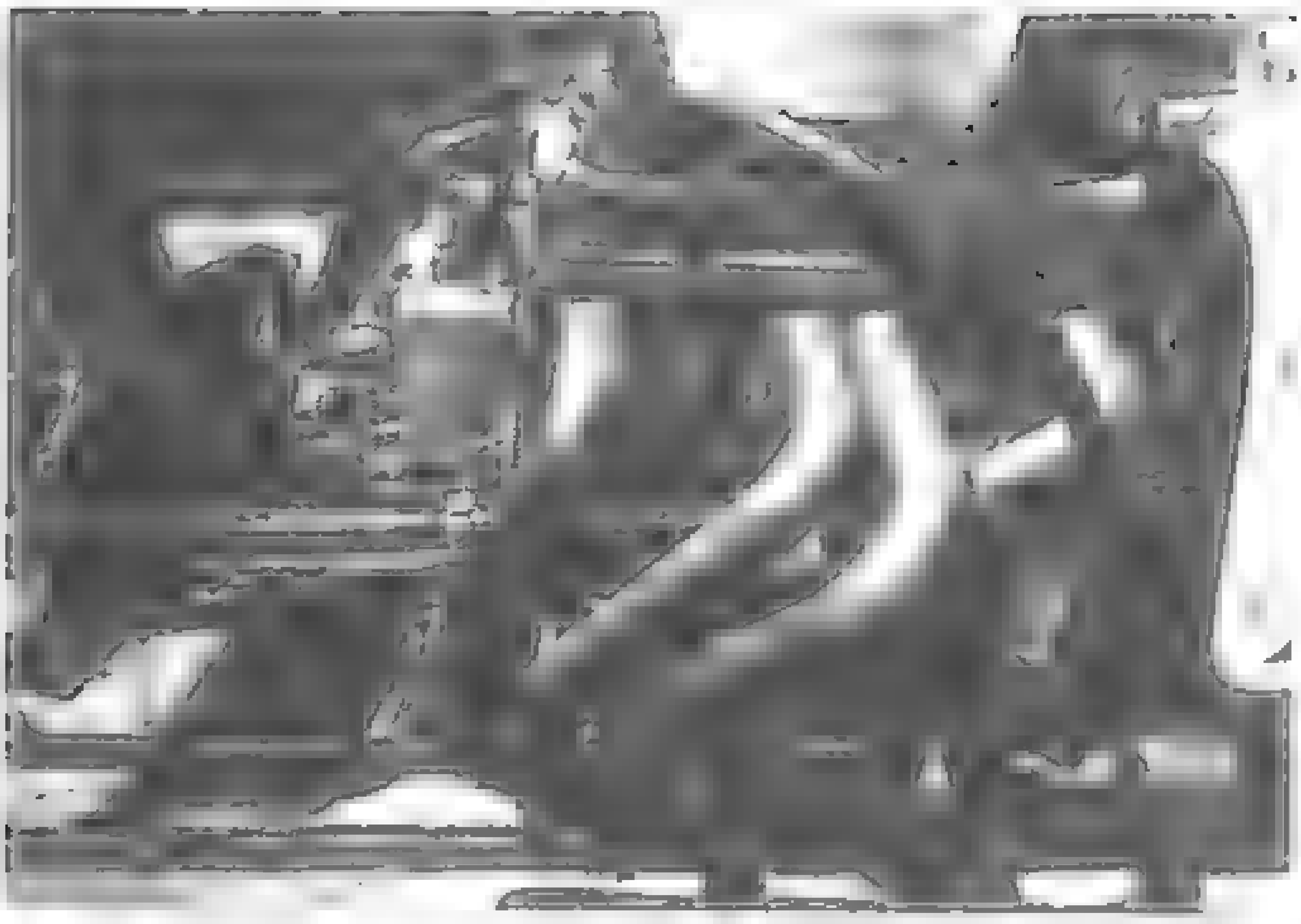
[illegible] anticipated, and a blown engine can result because nobody realised that an unscheduled top-up was needed.

However, it is during dyno development that a 'sighted' oil tank is invaluable [illegible] up to a maximum of 1¾ litres. This should be followed up with an on-circuit test [illegible] engine while simultaneously de-clutching. Coast back into the pits and drain the oil [illegible] testing, it indicates that the pick-ups are located incorrectly.

Mounting the oil tank in the engine compartment as close to the engine as possible cuts down the length of the various oil lines, and thus reduces flow losses. However, flow losses also occur when hoses are too small in diameter or are bent too tightly. The minimum radius of bends should be 2in; if you do not know what that looks like, take a length of 4in exhaust pipe. That has a 2in radius, so no bend should [illegible] the engine and whether they are scavenge or pressure lines. A typical 2-litre race engine will have an oil flow rate of around 15 litres per minute, while a 6-litre V8 will be double that. Thus while a 2-litre could run 1/2in bore pressure hoses and 5/8in bore scavenge and suction hoses, bigger V6s and V8s would need 5/8in pressure lines and 3/4in scavenge and suction lines.

ENGINE BREATHERS AND CRANKCASE VACUUM

With a dry sump system installed, large engine breathers are not required. With small engines a 3/8 or 1/2in hose from the cam cover to the breather catch tank is ample, while for bigger V6 and V8 engines run two hoses. There are two schools of thought here: some tuners advocate running an 'open' breather, one that vents directly to the catch tank, while others prefer a 'closed' system, which incorporates a check valve between the engine and the catch tank. Personally I prefer a closed system as this allows the scavenge pumps to run down the crankcase pressure very low, perhaps below zero when there is minimal blow-by. With zero crankcase pressure hp rises a little and oil leaks around the crank and cam seals, for example, dry up. If the engine runs sour during a race and blow-by increases crankcase pressure beyond what the scavenge stages can handle, the breather pipe flapper valve opens to vent excess blow-by back to the catch tank. With an open breather, pressure in the crankcase will never drop down to zero because if the scavenge stages have excess pulling capacity beyond



The pan evacuator system uses exhaust pulses to draw a vacuum on the crankcase. A good system will pull up to 30in w... vacuum, which is useful but a lot less than that possible w... engine pump. The ... evacuator cannot be used on turbo engines. The fitting into the exhaust pipe is an ...

...lling blow-by, they draw air into the engine via the breather pipe. In ... system this cannot happen be... check valve allows only one-way flow in the breather pipe.

In recent years the lub... en carefully scrutinised in the search for more power. It should be obvious ... a crankshaft rotating at 8,000rpm and banging into the gallons of oil draining from the camshaft, lifters and rockers, etc. will be experiencing a certain degree of drag – maybe not as much as you experience when you drive into a puddle of water at high speed, but there will be drag none t... This drag consumes a lot of power and results in oil frothing ... to the friction involved and energy being expended. 'V'-type push rod eng... offenders in this respect as they dump all the oil from the cam and l... the crankshaft.

You learned during school physics lessons that fast moving air creates a low pressure zone, which is why aerop... kshaft spinning at 8,000rpm is also moving the air at qui... s a low-pressure region in the area in which the crank and rods are moving. The rest of the cran...

...ur engine. Let us assume that there is a difference in pressure of 5 between the high-pressure and low-pressure zones in this engine. What will tend to happen is that the oil pour...

crankshaft, because the centrifugal force of the crank is unable to overcome the pressure differential within the crankcase. Of course part of this mass of oil must [illegible]

caused by oil drain-back from the cam and lifters.

An effective dry sump system can help with this problem in the following way. If scavenge pumps are used, [illegible]

in dry sump engines.

The other reason for the rise in hp is due to improved piston ring to cylinder wall [illegible] compression ring, we get a superior ring to cylinder wall seal. This comes about [illegible] ring and the bottom of the ring groove. If we achieve zero leakage here more combustion pressure builds behind the ring forcing it hard against the bore to effect a more gas-tight seal. Obviously with added combustion pressure trapped above the [illegible]

There are limits to the amount of vacuum desirable in the crankcase. First off it costs us power to drive a scavenge pump harder, so we reach a point of diminishing returns. [illegible] vacuum can pull so much oil away that piston pins seize unless we pressure lube them using special rods with an oil passage from the big end to the little end. Exhaust valves [illegible] scavenge pump reliability problems. Consequently my recommendation is not to exceed depressions of 10–12in for street engines. For competition engines start out at [illegible] Honda RA122E/B started out running modest crankcase pressure of 64% of atmospheric. The vacuum was later increased to lower pressure to 30% of atmospheric, and maximum power rose by 16hp to 774hp at 14,000rpm.

SCREENS AND SCRAPERS

The dry sump system cannot get all the oil away from the crank and for this reason we need a system of screens and scrapers to assist (Figure 12.5). The screen should be of [illegible] as shown, in close proximity with it. The idea is that the screen will prevent oil from splashing back on to the crank after it has been flung to the floor and sides of the oil [illegible] there is less energy available for splash-back when the oil hits the pan.

A rigid 22 gauge steel scraper positioned along the side of the oil pan deck will scrape oil from the crank and rods if machined to extend to within just a few thousandths of an inch of these.

To improve the oil drain off from the scraper, the side of the oil pan should be

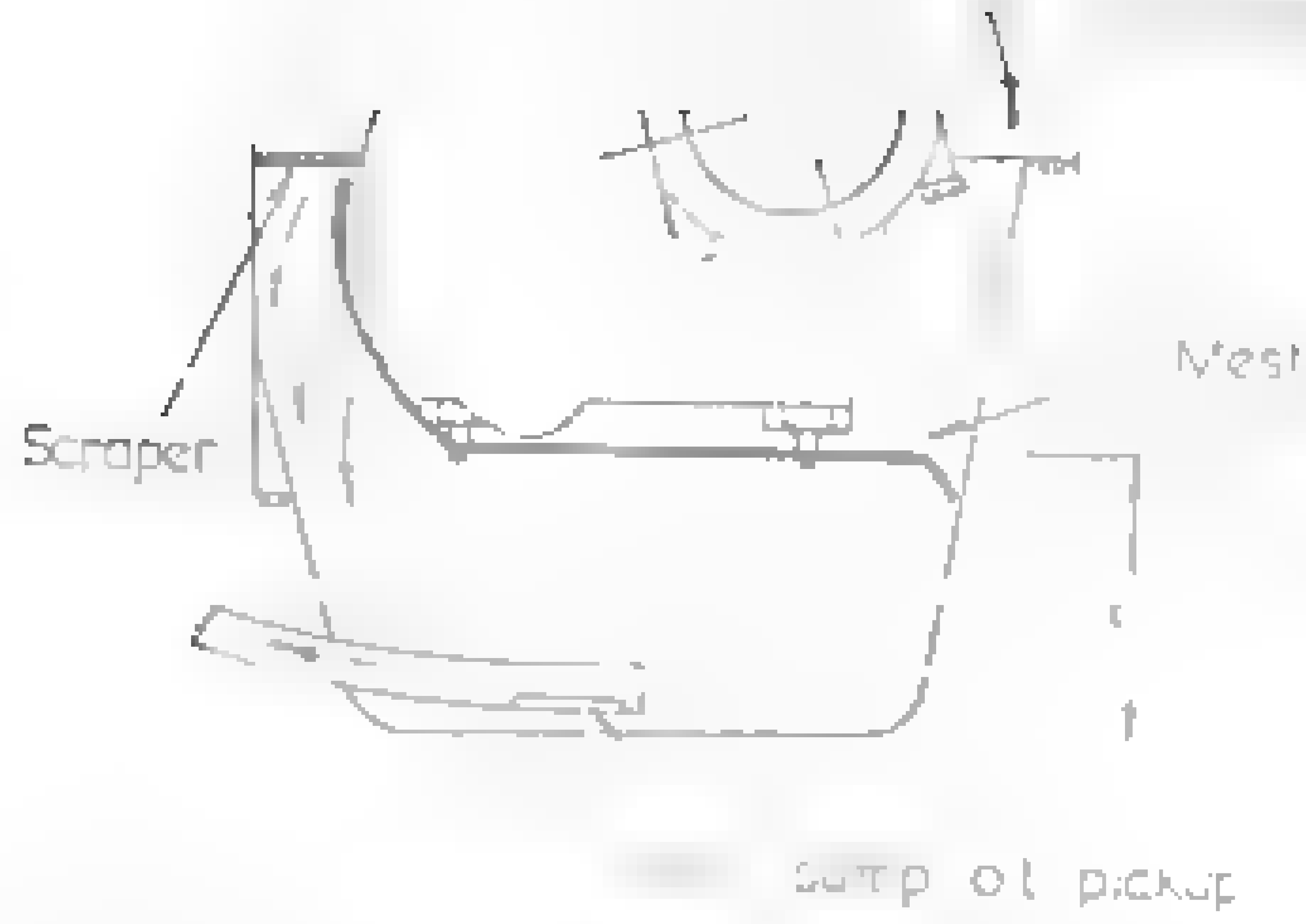
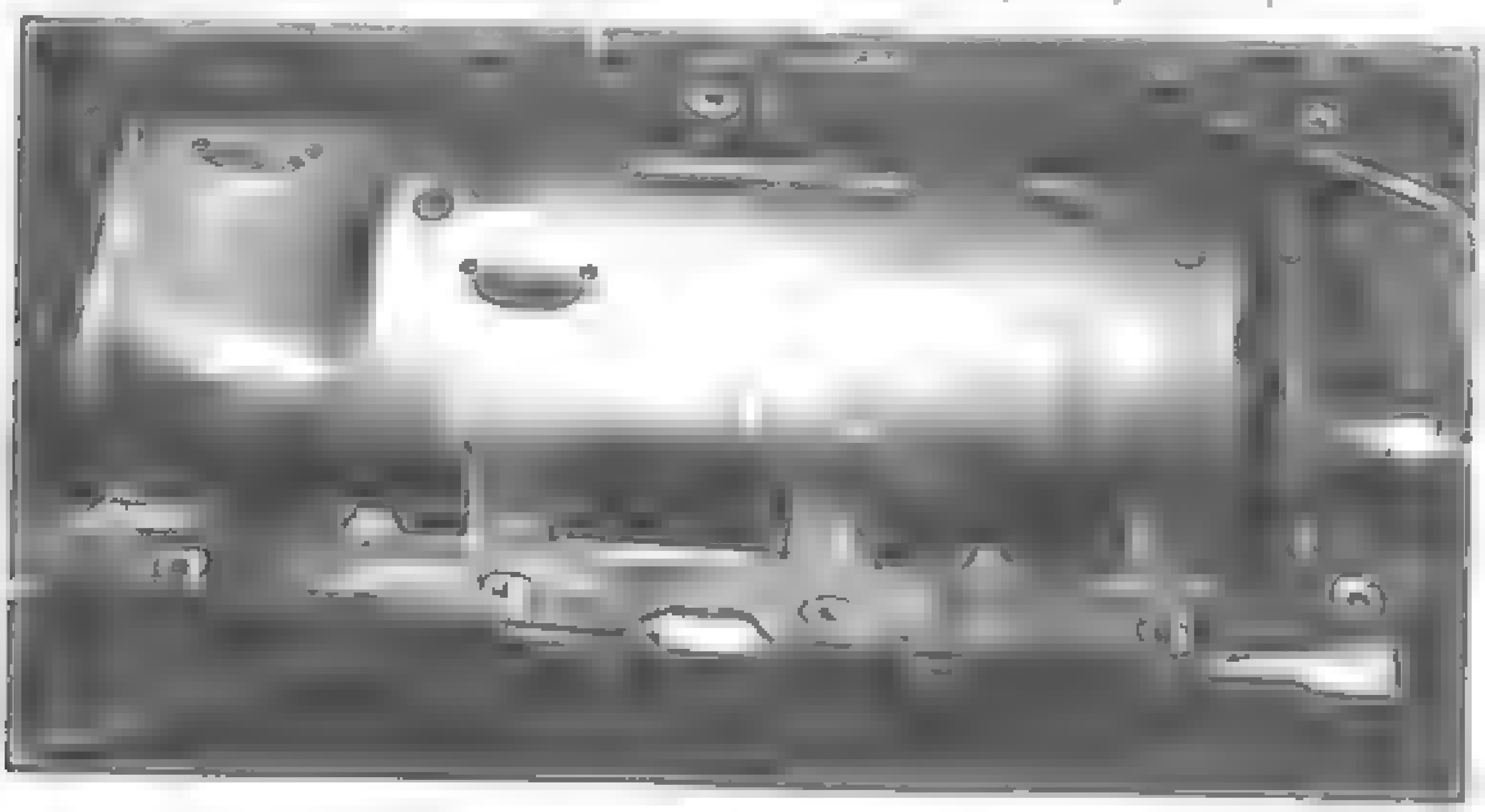


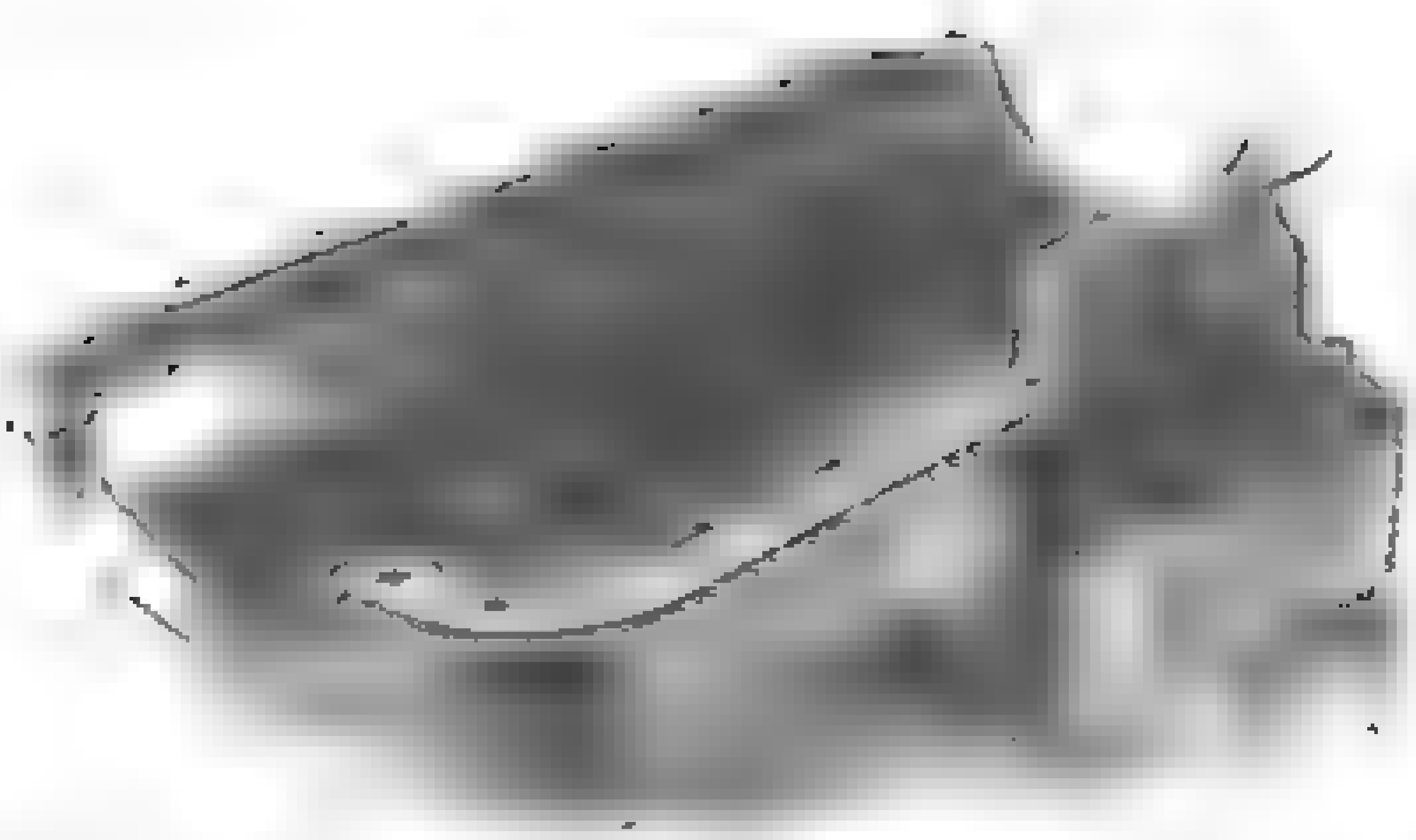
Figure 12.5 Oil screen and scraper

extended out an inch if possible. With these modifications you can expect a further 3–4% power increase over and above that possible with dry sumping alone.

Do not despair if you cannot afford a dry sump system, because the screen and scraper set-up can be adapted to work almost as effectively with a wet sump. There are two ways to approach the problem. If ground clearance is not a consideration, deepen the sump to give 1½in clearance between the screen and the horizontal anti-surge baffle. For improved oil drain from the scraper, move out the side of the sump 1in, the same as for the dry sump oil pan.

The windage tray and scraper strips oil off the crank and rods and prevent the low pressure area around the spinning crank from drawing oil up out of the sump





This sump has been widened to lower the oil level. Additionally, a scraper and both a horizontal and vertical baffle have been added

If ground clearance is a problem, the only alternative is to widen the sump to lower the oil level. Again there should be $1\frac{1}{2}$–2in clearance between the screen and the horizontal baffle

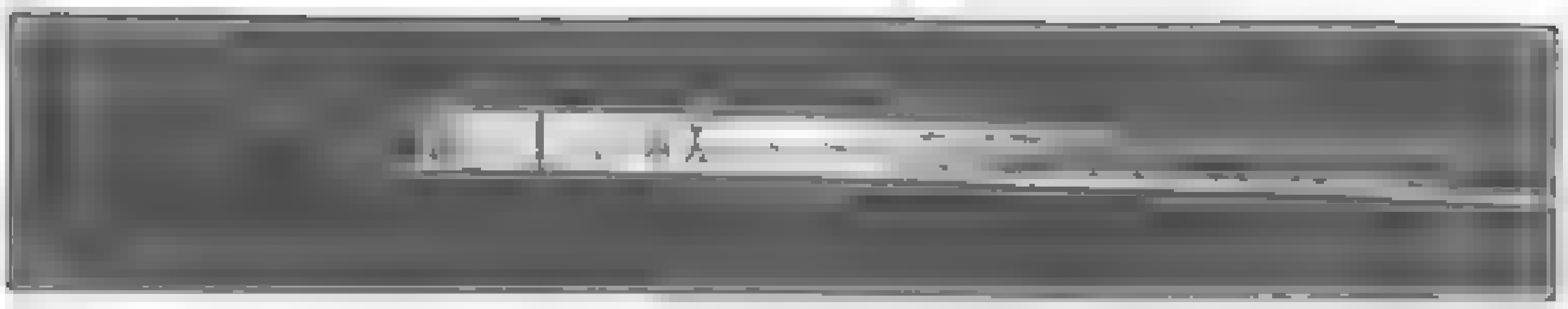
OIL FILL LEVEL

Having considered the foregoing information it should be obvious that the sump should never be overfilled, otherwise you are sure to have the crank and rods dipping into the oil every revolution. If that is not bad enough, you will end up with a situation where too much oil is being thrown up the cylinders; the rings have to drag it off the walls and that consumes power. If the rings cannot cope with this additional oil load you will have combustion chamber and plug oiling problems, which will affect combustion and rob you of performance

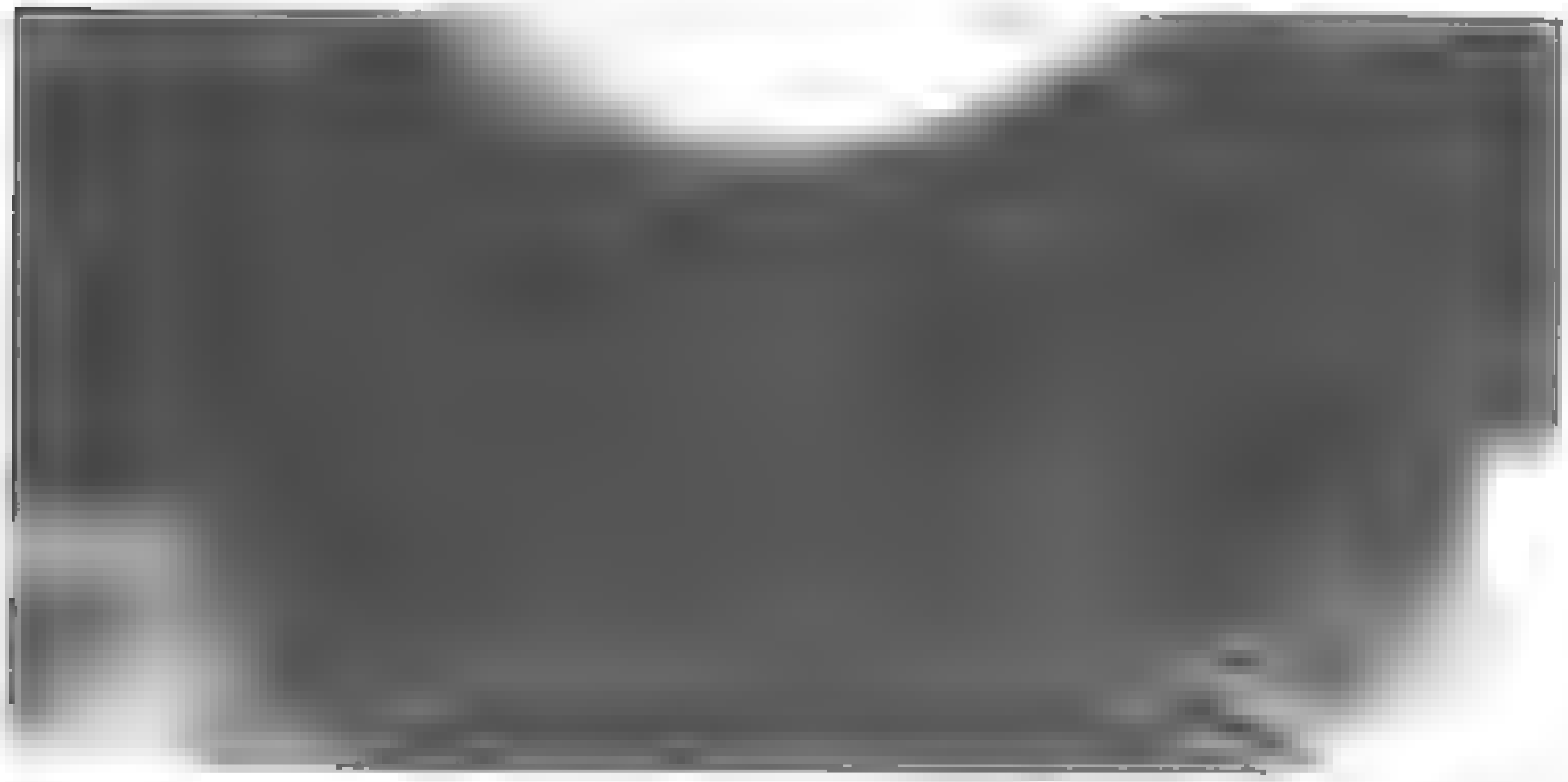
Personally I keep the oil level well below the full mark on the dipstick. First I check for oil surge with $\frac{1}{2}$ litre less. Then if there is no surge problem I drop another $\frac{1}{4}$ litre and so on down to a maximum of 1 litre below full. With a complex baffling system involving several swinging trap doors and a swinging pick up I have been able to drop to a 50% oil fill This is dangerous territory though, and the hp gains are minimal. Lowered 1 litre some engines have shown maximum hp gains of around $2\frac{1}{2}$%, but a little over $1\frac{1}{2}$% is more usual. In road cars at cruise the gains are much greater, and this pays a nice fuel economy dividend

CRANKCASE AIR PASSAGES

We may have an opportunity to go much farther in our endeavours to reduce crankcase windage losses. For example in the upper echelons of motor sport, engines are built with the crankcase divided into separate compartments so as to limit air movement and subsequent frictional losses caused when a piston rapidly descends in its cylinder Obviously if an engine has an individual cylinder capacity of 500cc each time a piston descends it has to push 500cc of air out of the way If it is only a 4-cylinder engine working at 7,000rpm we are pushing the equivalent of 7,000 litres of air back and forth



To reduce parasitic losses, and ... level mark. If possible run closer to the 'min' mark



Above: These passages between the cylind... air below the pistons to easily shut... rth as the pistons go up and d... properly ground out they partly blo...

Below: Using a grinder all corners are radiused to help prevent main bearing web cracking, plus air flow is improved, cutting pa...

in the crankcase every minute! That costs us power even if we have a slight vacuum in there. Remember, normal air pressure is 14.7psi, so if we apply a vacuum of 15in, about 7.4psi, we are still moving a lot of air (about the equivalent of 3,480 litres). When we divide the crankcase into compartments we arrange things such that the air being displaced as the No 1 piston descends moves next door to fill the void created as the No 2 piston rises. In this manner air movement is restricted to a smaller area, windage losses decrease and hp rises.

Car engine manufacturers have done something similar. In the late '80s I noticed that some engines began appearing with a large passageway between the main bearing webs and the bottom of the cylinder bores. Some tuners suspected this was to save engine weight. However, those who were observant also noted that at the same time these passages appeared the very same engines were outfitted with windage trays which wrapped up close to the crankshaft. While these pressed metal trays did a reasonable job of pulling oil off the crank and rods, and prevented the hurricane being generated by the spinning crank from drawing oil up out of the sump, they created a secondary windage problem. There was now no easy path for the air to shuttle between cylinders as the pistons travelled up and down. The solution was to add these holes to interconnect the bottom of the cylinders. The resultant windage losses produced hp gains in the region of 2%, and in a few engines it was over 2½%.

In some blocks these holes are machined, but generally they are cast. Either way they need some help to properly do their job, and at the same time we can partly get rid of a potential failure point in the block. We have to view these passages in a similar light as the inlet ports. They are expected to move the same gas volumes, but in considerably less time. Cast passages are usually quite large. However they are often partially blocked by casting slag. This stuff has to be ground out to improve flow capacity, and more importantly to get rid of a stress raiser which could end in the main bearing web breaking away from the bottom of the block. Machined passages are better in this respect, however, as they are quite small be sure to radius both ends of the hole to help flow.

WET SUMP BREATHERS

To vent crankcase pressure most engines in standard tune rely on a single ½ or ⅝in hose connected to the air filter or inlet manifold. Any wet sump competition engine should have at least a 1in hose connected into each rocker cover and another 1in hose connected into the block. Take care when connecting breathers into the motor that oil is not going to be splashed up the breather pipe by a rocker arm or a con rod. If such is the case, steps will have to be taken to fit a suitable deflector at the breather inlet.

The problem with these breather systems is that they fall far short of what we actually want to help us make more hp, and improve engine reliability. The stock breather system is required by law on road cars to cut pollution of the air we all breathe. We aren't allowed to simply dump blow-by gases into the atmosphere as is done on competition engines. Thus this gas, along with any oil mist that the oil/air separator in the cam cover has failed to remove, is recirculated into the inlet tract. Clearly this gas occupies volume and so reduces the power potential of the engine. The oil mist though can wreck engines by lowering the fuel octane and setting off detonation.

CRANKCASE VACUUM PUMP

How can we get around this and still do our part in not ruining the environment? In the past I have used the belt driven smog pump that was fitted on some engines back in the '70s and '80s to pump ... the exhaust ... lower emissions. However ... than have the pump ... I connected it to suck blow-by ... an oil/air separator tank which I connected to the crankcase and cam cov... pumps are quite reliable if not spun to ... protected from exhaust ... by a check valve. ... the blow-by gas into the e... single stainless st... the exhaust at the exhaust ... Obviously in turbo engines these ... must be recirculated after the tur... the dump pipe or just prior ...

The main problem with ... smog pumps, properly called an AIR or Air Injection Reactor pump, is ... (5in dia x 3-4in long) and didn't pul...

...ed two dedicated vacuum pumps which are very effective. The big 4 vane ... draw up to about 12in. The 3 vane is a better street pump, but is also useful in endurance events where you are required to run a wet sump. The 4 vane race p... is enhanced if some oil mist is allowed to flow through them. This means that it is best to locate the oil/air separator after the pump, and in road cars with a conn... the exhaust the back fire check valve ... after the separator. Street engines routinely show a 3–4% hp rise at 10–12in vacuum – over double of what I record with smog pumps.

... sump engines a Moroso vacuum pump can be used to improve piston ...ng and thus reduce blow-by and oil contamination, plus gain hp

Chapter 13

Cooling

Y or indirectly on a flow of air to stabilise the temperature of the cylinder head and the cylinders. The cooling system of every internal combustion engine performs a vital function: the dissipation of heat in order to maintain normal engine operation

The heat engine relies on the conversion of fuel into heat, and then into mechanical energy to produce power at the crankshaft Only about one-third of this heat is converted into power, another third is eliminated through the cooling system Right away this should alert you to the load placed on your cooling system. Thus if our engine is generating 300hp we are also calling on the cooling system to get rid of the equivalent of 100hp of heat energy. Now if our stock engine produced 150hp the manufacturer obviously installed a cooling system capable of handling that heat load and not much more, so it would be foolish of us to think that the stock arrangement could cope with 300hp

AIR-COOLING

If your engine is air-cooled, there is not a great deal that you can do to increase the engine-cooling capacity to cope with higher power outputs. Therefore it is essential to ensure that the cooling system provided by the vehicle's manufacturer is operating at 100% capacity.

Heat radiation from the cooling fins is retarded by the presence of oil and mud, so ensure that the fins are clear Fins that are silver-coloured can have their radiating capacity improved by a coat of matt black paint

Anything that is obstructing air flow on to the head and cylinders should, if possible, be moved to another location. I am amazed by the number of bikers who persist in fixing lights, air horns and oil coolers in front of the engine – the idea is to encourage air flow over the motor, not restrict

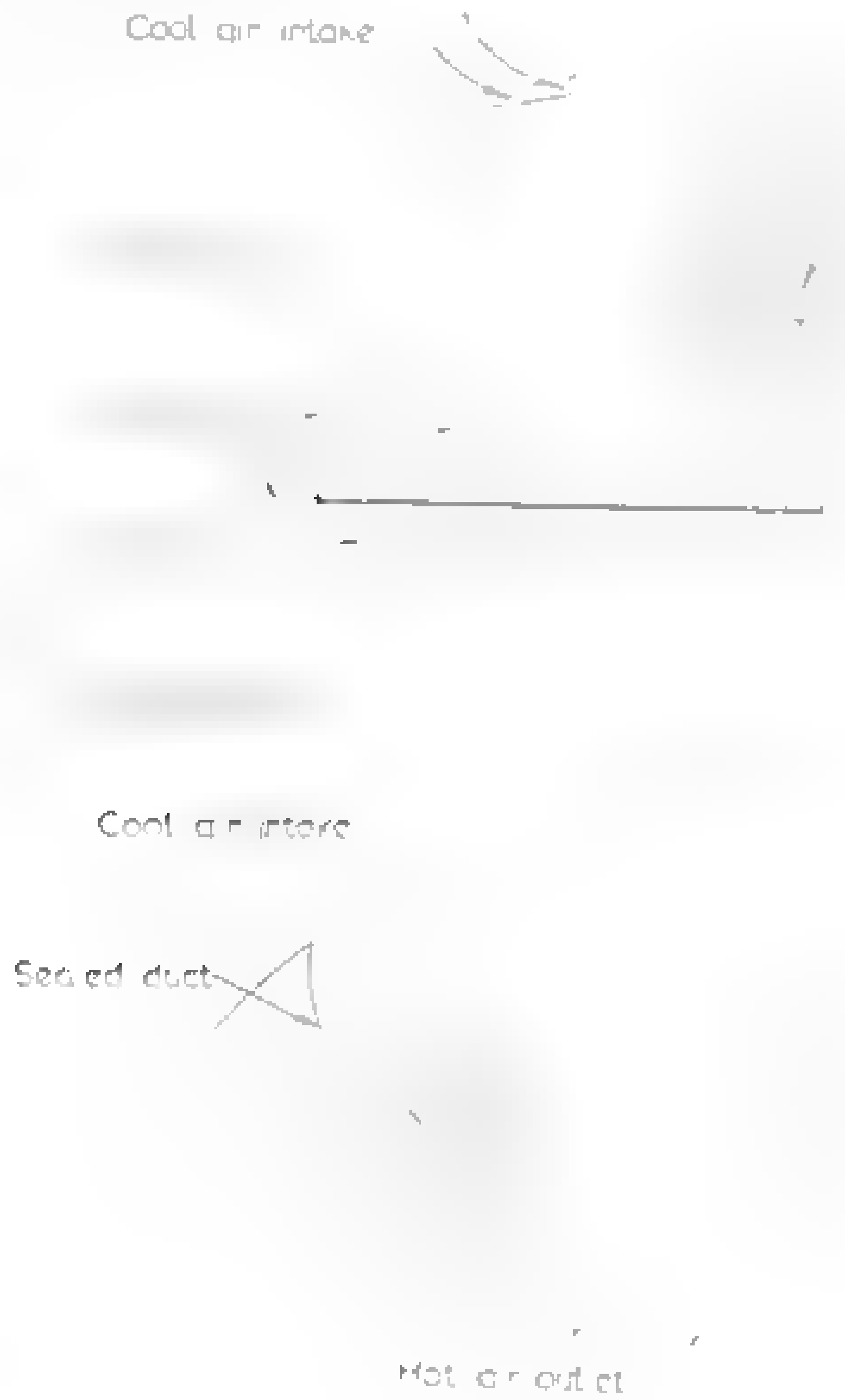
Air-cooled cars rely on a fan to circulate cooling air over the head and cylinder Think carefully before you decide to modify a cooling system of this type, as there are

several arrangements in service from the various manufacturers. Without a proper understanding of the system any modification could spell disaster for the engine.

The first consideration is whether the air inlet or outlet is ducted, because it is important to keep the inlet and outlet separated. If the inlet is taking in hot air that has just been expelled through the outlet, overheating and even a blow-up could result.

On the VW Beetle the outlet is ducted (Figure 13.1). The fan intake is open inside the engine compartment and the hot air is exhausted beneath the engine tray. [illegible] so ensure that all the tinware is properly fitted and sealed with Silastic. Check the rubbers around the spark plug caps, hot air escaping through here will be re-circulated. Badly planned body modifications may also allow the entry of hot air into the engine compartment, so look into this as well.

Figure 13.1 VW cooling systems



By contrast, the Type 3 and 4 VW use ... The outlet is open and ...

the inlet duct remains sealed and that any modifications d... t bring the inlet an outlet so close that hot air will be re-circulate...

Most air cooled cars have their air intake in ... area of low air pressure, which reduces the amount of air available for cooling. Never fit a different pulley to slow the fan. Some claim that this gives better cooling because the fan is more efficient at lo... speeds. Also they feel that because less power is being used to drive the fan performance will increase. I have found the opposite to be true ... slow up the f... and ...low up the engine.

It may be possible to increase fan speed to increase air flow over the engine ... the other hand the fan may al...ady be moving all the air it is capable of movin... any increase in engine rpm or fan speed wil... ...ginal flow improvement. Frequently, increased cooling can be best cared for by thinking more in terms of massively increasing the size of oil coolers, and perhaps upping oil flow in the engine. Especially, there is a need for more flow volume up into the pistons and ...s the cylinder heads.

When Porsche added turbos to their 630hp 917 racer, increasing the output to 1,000hp and then later to 1,100hp, they successfully increased fan speed from ... of 7,400rpm up to 9,000rpm, a 22% increase. However cooling air flow rose fr... ...0ltr/sec to 3,100ltr/sec indicating that the big fan on the 5ltr flat-12 (5.4ltr on later version) was more efficient at this higher speed. Also you should no... that, due to the much higher hp, full throttle was used less frequently and fo... significantly shorter periods. As a result air flow per hp was droppe... from 3.8ltr/sec to ...

...f air-cooled engines can be improved by ensuring only cool air is being draw... also possibly by increasing fan speed, and by increased oil cooling, ie increasi... to the pistons and cylinder heads and larger oil coolers.

the naturally aspirated engine to 2.8ltr sec in the 1,100hp turbo engine without any problems with overheating.

MONITOR CYLINDER HEAD TEMPERATURE

Before you even think about modifying a fan air-cooled engine, be sure to fit a cylinder head temperature gauge; it could save you a lot of money. The liquid surrounding the combustion chamber of a water-cooled motor tends to act as a heat [illegible] little in reserve to take away a sudden increase in heat, so the combustion chamber and [illegible] melt the pistons. For this reason the head temperature must be continually monitored when you are engaging in high speed motoring.

What is the maximum safe head temperature? That depends on where you measure it and on the quality of the pistons. If pistons suitable for competition are being used and the temperature is being recorded from a thermocouple washer at the base of the spark plug, I would suggest throttling back once the temperature exceeds 220°C. If the thermocouple is attached elsewhere on the head, I would advise that the same action be taken at a 200°C read-out.

LIQUID-COOLED ENGINES – CORROSION AND FREEZE PROTECTION

Cooling by water or liquid is usually thought to be more straightforward than by air, and to a degree this is so. For instance, it is easy to accommodate a large power increase from a liquid-cooled motor and avoid overheating; you simply use a larger radiator. Other aspects of water cooling are, however, not so elementary.

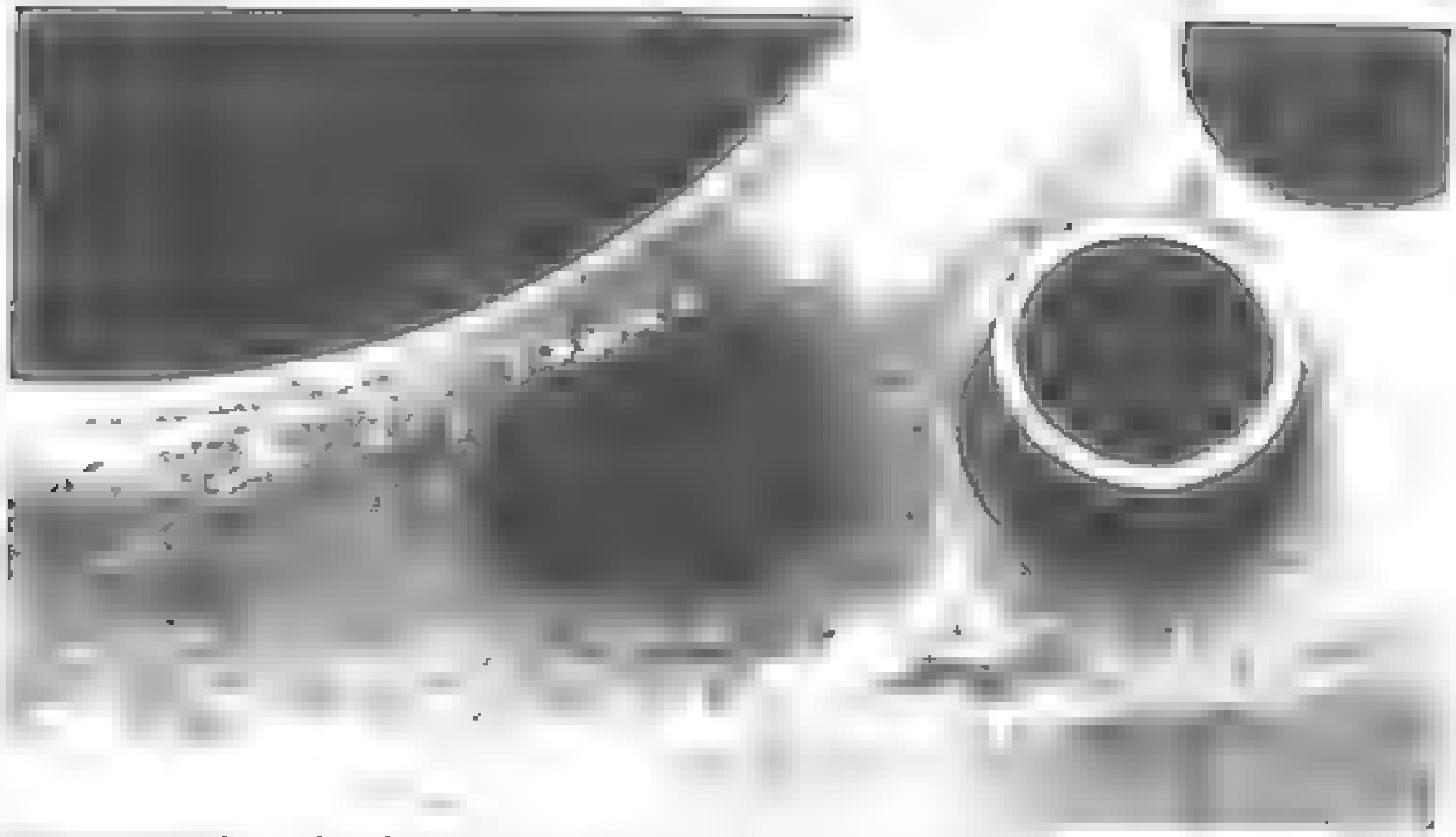
The two major deterrents to proper heat transfer from the combustion chamber and [illegible] cooling system.

Metallic oxides (eg iron oxide, or rust) are formed in the water passages. A [illegible] in order to maintain optimum heat transfer within the engine, the cooling system should be [illegible] inhibitor that will keep water jacket surfaces clean and free of deposits.

There are two basic types of inhibitors: chromates and non-chromates. Sodium [illegible]

Non-chromate inhibitors (borates, nitrates, nitrites) provide protection in either water only or water and permanent anti-freeze systems. Chromates must not be used in systems protected by anti-freeze.

[illegible]



lew a head gasket due to corroded water passages. The glycol coolant
t, because of a lack of co enance, had in fact turn

orrosion an l minera t jackets, a mixture c l distilled water
r or a mixture of distil ed water treeze should be used, and
l as recommende l

Anti freeze containing cooling system sealer additives must also not be used due to the possibility of the sealer plugging the radiator core tubes and various other areas in the cooling system. In fact, a sealer or stop-leak of any description is not to be [illegible], except in an emergency to get you home or to finish the race. Then, as soon as possible, it should be flushed out of the cooling system by a cooling specialist using high-pressure air and water

[illegible] clear of radiator sealers. Soluble oil turns water a milky colour when added

ELIMINATING AIR BUBBLES

The presence of air bubbles in the coolant reduces its heat transfer capacity and the efficiency of the water pump. Air can be sucked into the system through a leaking hose or gasket, and gas bubbles can form in the system due to localised boiling, or due to water pump cavitation

In the first instance air can be kept out by ensuring that the system is free of air or water leaks, and by maintaining the coolant at the correct level. However, a good deal more is involved. At the very least take care to bleed the cooling system carefully whenever it is disturbed in any way to replace a hose, etc. You must ensure that there are no pockets of air trapped in the block and head, or, in the case of road cars, in the interior heater. Some engines will self-bleed fairly satisfactorily, while others may have to be bled through a factory fitted bleed nipple. Others, however, require the addition of a bleeder nipple high up on the block right under the deck and/or a bleeder in a high point in the head

After letting trapped air escape through the bleeders, do not assume that the cooling system is now free of air. With a race engine running a restrictor, start the engine and throttle it up and down for a couple of minutes. The vibration and water surge should get most air bubbles moving to the high spots, so then open the bleeders to release any air and top up the radiator

A road car is a bit different as we also have to worry about getting air out of the heater and from under the thermostat. What we need to do after initially bleeding the system is to take the car for a run for about 15 minutes with the heater full on. The thermostat will open in this time and the combination of engine vibration and water surge in the passages should get any trapped air moving either into the high point in the head or into the radiator. At the end of the run, with the engine still running, carefully open the top bleeder to let any air escape. With that done, shut off the engine and allow it to cool. When down to ambient, check the coolant level and top up if necessary

Even after all these steps, the cooling system will probably contain some pockets of trapped air that will only be eliminated by evacuating the system using a vacuum pump. People accept the fact that refrigeration and air-conditioning systems must be evacuated prior to being re-gassed, and in reality race car cooling systems deserve the same treatment. What is required is to pump all the air out of the cooling system and bring the vacuum down to around 25 inches of Hg. After turning the vacuum pump

off, the vacuum should hold at that figure, if it does not there is a leak somewhere that has to be eliminated. With all leaks rectified the system can then be completely filled with coolant, free of any air pockets.

THE PRESSURISED COOLING SYSTEM

[illegible] is that we pressurise the cooling system. The boiling point of water is 100°C (at sea level), so why is it necessary to increase the boiling point by pressurising the system [illegible] even lower temperature, usually 70°–75°C?

First, the system is pressurised to prevent boiling after the engine is turned off. Once the coolant stops circulating, its temperature climbs rapidly from its normal [illegible] (14.7psi). If the water boiled each time the engine was stopped, a considerable amount of coolant would be lost, and if the system was not re-bled an air pocket would form.

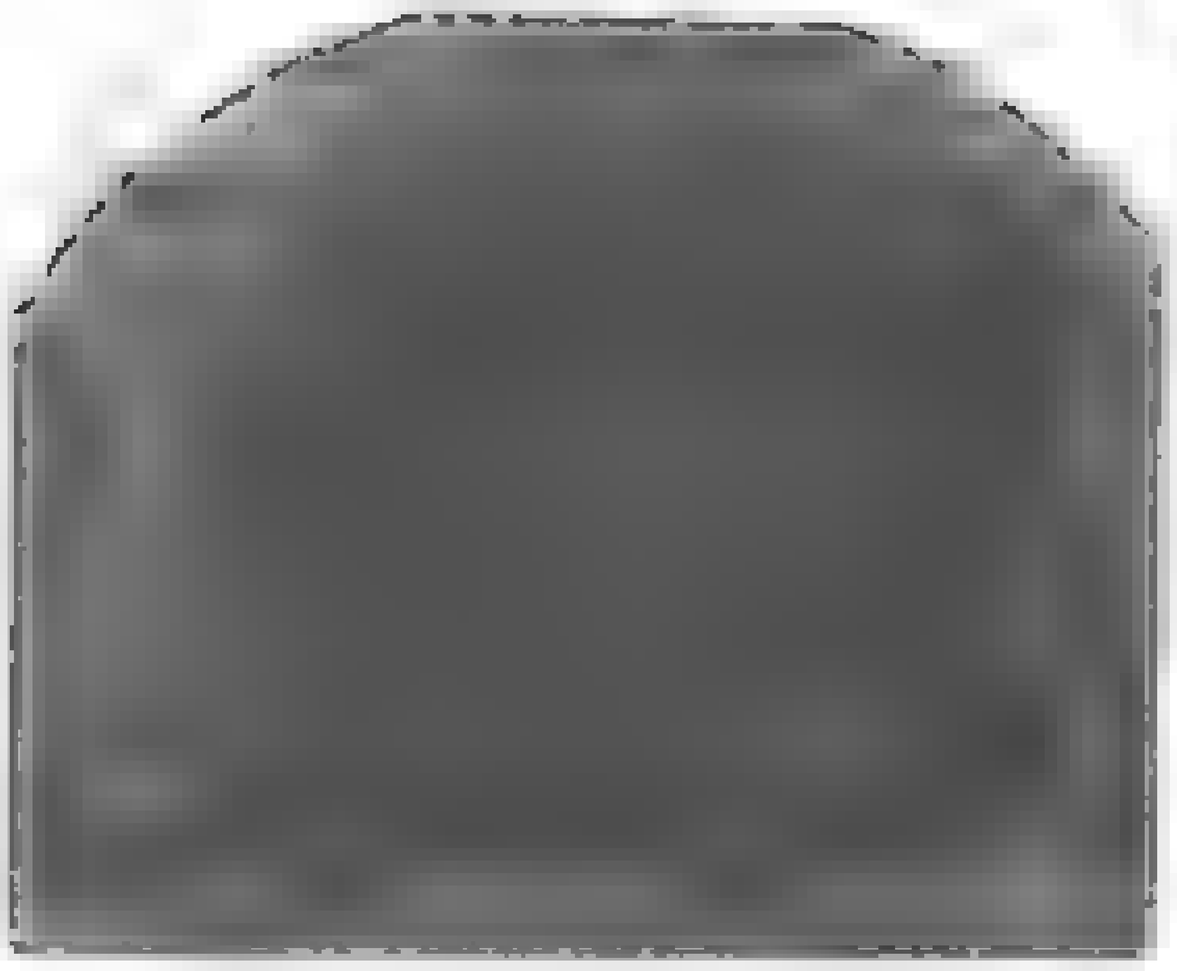
Second, regardless of what the temperature gauge is reading, the temperature is very high in the water passage around the combustion chamber, particularly close to the exhaust valve. Remember that the temperature gauge is only giving a reading of the circulating water temperature, not the temperature of the water around the exhaust valve seat, where the temperature is well above the boiling point of water. To prevent the water around the combustion chamber from boiling and forming a steam pocket, the cooling system has to be pressurised. If the coolant were allowed to boil here, localised heating of the metal would occur, creating thermal stress points that would lead to cracking of the metal.

By pressuring the system using a 14psi radiator cap the boiling point is raised to approximately 125°C at sea level. As well as preventing boiling when the engine is switched off, the radiator pressure cap also serves to stop gas bubbles in a number of other situations. For example, a road engine may be given a lot of throttle at low engine speed, which will give rise to rapid heating of the combustion chamber, exhaust port and valve area. At low engine rpm the water pump is turning slowly so the water flow is limited and the water pump will not be creating any pressure in the block and head. It is only the radiator cap that stops this sort of localised boiling.

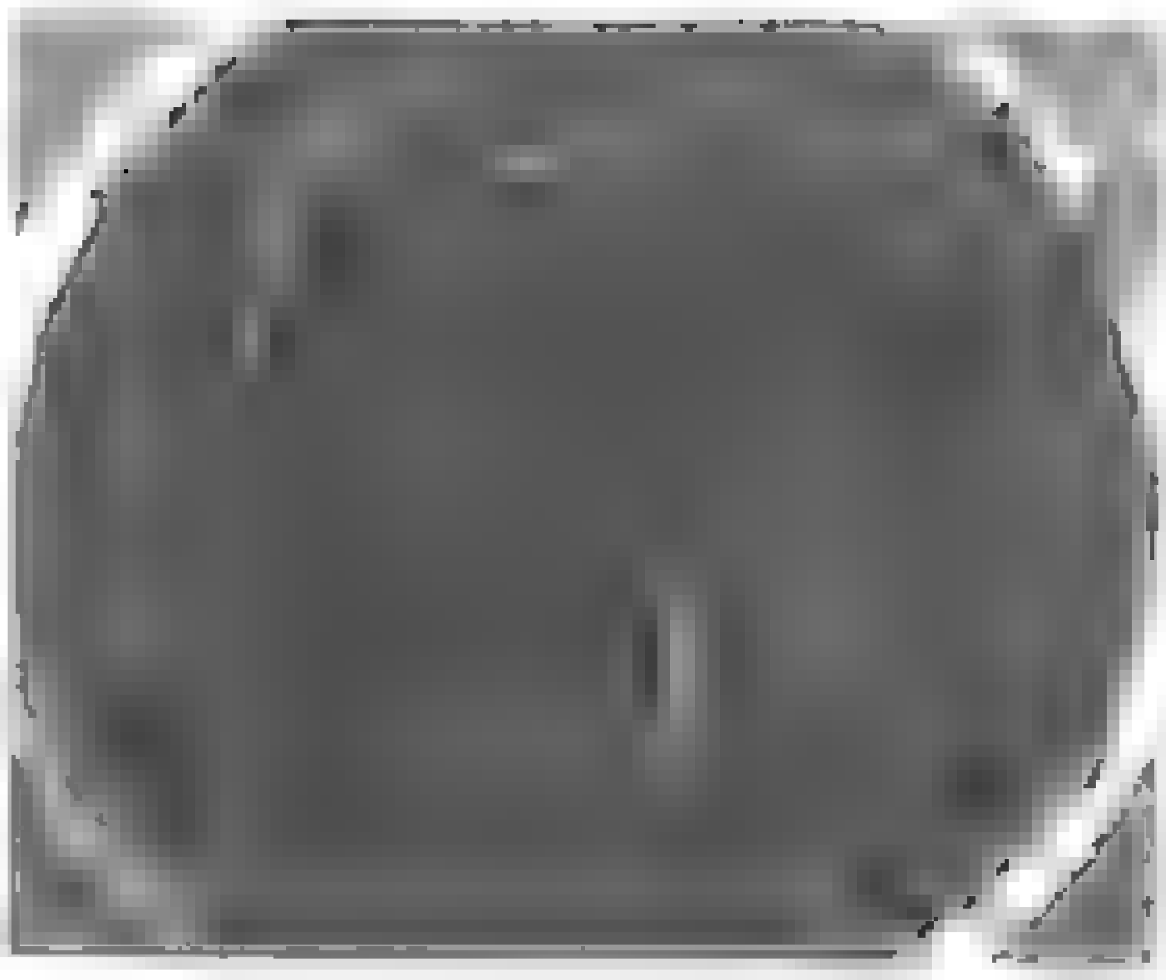
A similar sort of thing occurs when a race car pits. The top of the engine is extremely hot because of all the full-throttle driving out on the circuit, but now, with the engine idling while adjustments are carried out, tyres changed, etc, the engine would boil without a pressure radiator cap.

WATER PUMP CONSIDERATIONS

However, when water pump speed increases to the peak efficiency speed of 4,000–6,000rpm it is not the radiator cap that stops boiling but water pressure created by the water pump. Even many race engine tuners do not seem to understand this: regardless of the radiator cap pressure a water pump spinning at maximum efficiency rpm will produce a pressure head of around 30–40psi in the engine block and head when water flow out of the head is limited by a thermostat or restrictor plate. This



An open back impeller is inefficient, and consequently this water pump has poor pumping capacity and a strong t... cy to ... ate even ... relative ... ars to be fast on the impeller ... erosion due to cavitation



A closed back impeller is much more efficient and far less prone to ca... The four holes promote ... lubricate the sea...

pressure packs coolant around the top of the cylinders and around the combustion chambers to carry away combustion ... n insulating ... ket of gas bubbles forming in these areas.

In the third situation, gas bubbles due to water pump cavitation have to be tack... on two fronts: water pump des... nd water pump rotation speed. Ever... pumps cavitate if you spin ... t some stock pumps ... n to form bubbles at relatively low rpm due to both poor design and cost-cutting manufacturing techniques. In the past, water pump impellers were cast with a closed back ... clearance between the pump body and impeller was kept ... to ensure maximum flow volume. However, in modern times stock pump impellers tend to be pressed from light metal, which results in an open-back impeller often ... ed well out from the pump body. This design will not only pump less volume, which also means reduced block pressure, but it will also begin to bubble the water at low ... pump speeds.

Race pumps ... available for some of the more popular engines, look f... fitting ... ture a stronger ... it, but with ... body and impeller flow vol... sistance may be no better than the stock pump.

If no good race pump is available you will have to modify the stock pump. It may be possible to modify a closed-back ... impeller from another pump, which may involve searching out old engines in sc... An alternative is to close up the ... t an open impeller by welding a backing plate on to it. Then, when the impeller ... back on to the pump shaft, press it on f... ner to reduce the pump ... impeller-vane clearance.

With that sorted out we now have to turn our attention to pump speed. In a well designed pump, water flow will increase proportionally to pump rpm up to about 5,000 rpm. Between 5,000 and 7,000rpm water flow continues to inc...

of increase begins to taper off rapidly as the water begins to bounce around off the impeller vanes and pump body. Thus past a pump speed of about 5,000rpm it is costing more hp to move proportionally less water. In a 650hp race engine a water pump running at 5,000rpm [illegible] and the parasitic loss increases to 25hp for only a 15–20% increase in water flow.

As pump speed increases the pump will begin to cavitate, at which point not only a lot of air [illegible] temperature and block pressure also influence at what point cavitation will set in. As the average system temperature increases past about 85°C, the pump rpm at which cavitation begins decreases. Likewise with block pressure, less block pressure means that cavitation will occur at a lower pump speed. In a race engine this means that we must limit water pump speed and control engine block water pressure, and not concern ourselves only with water temperature.

First I would look at pump speed. While most pumps tend to work at peak efficiency at 5,000rpm, they do vary, so check with the pump manufacturer or do some flow testing yourself. If the pump peaks at 5,000rpm and the engine produces maximum hp at 7,750rpm, I would fit a big pulley to the pump to underdrive it to 65% of engine speed. Thus at 8,500rpm, the engine redline, the pump will be turning at around 5,500rpm, and down at 6,000rpm it will be running at 3,900rpm. Basically I am aiming to keep the pump in its maximum efficiency zone of 4,000–6,000rpm.

CONTROLLING COOLANT FLOW RATES

Next I would look at restrictor size to control the rate of water flow through the engine and radiator, and to get water pressure in the block up around 30psi. Let us say that a restrictor with a 16mm hole is fitted and the block pressure is where I want it but the water temperature is too cold. The options are to go to a smaller restrictor, or a lower pump speed, or a combination of both. If the testing was being carried out in hot weather I would go for less pump speed to reduce parasitic losses, but I would keep an eye on block pressure [illegible] conditions a smaller restrictor would be fitted. If that brought the water temperature up, but there was a steep rise in block pressure to over 35psi, I would probably try a slightly slower pump speed combined with an intermediate size restrictor hole.

[illegible]

If going to a 19mm hole did not bring the temperature down and we were losing block pressure, we would have three options: a larger radiator, more pump speed, or a combination of both. Assuming that it was a hot day test I would go for more pump speed with a limit of around 6,200rpm. However, if the temperature was way up on a cold day, more pump speed might not get you out of trouble when the weather warmed up. Therefore a larger radiator might be the way to go.

Some of course would just go for bigger and bigger restrictors, or discard the restrictor altogether. This is not a good move, certainly the temperature gauge may come down to where you want it, but consider what is happening inside the engine. With no outlet restriction, water pressure in the block and head will be right down. With lowered water pressure steam pockets will form in the hottest areas of the engine. The combustion chamber, exhaust valve and piston crown will then overheat [illegible]

the engine into detonation. At this point hp will fall, and if the situation continues the engine will be destroyed.

An old wives' tale states that if you discard the thermostat or the restrictor the engine is damaged because the water is flowing through it too quickly to draw off excess heat, or it is flowing through the radiator too fast to give up its heat. This is not so; what causes the engine damage is insufficient water pressure to pack the coolant in tight around hot spots in the engine. Then any water that hits these hot spots dances about like water droplets on a sizzling barbecue plate without drawing off any heat. As the water boils off, the size of the hot spot grows as a bigger and bigger steam pocket forms.

In reality we want reasonably rapid water flow through an engine, as this tends to reduce the incidence of stagnant high-temperature pools. Additionally the rapid flow will scrub off gas bubbles as they appear in the hottest parts of the engine before they have a chance to congregate into an impenetrable steam pocket. In fact, the solution to cooling problems is not so much a matter of moving more water through an engine, as moving less more rapidly. This will pay large dividends in more hp and better engine reliability.

COOLANT FLOW PATH

[illegible] velocity to critical areas where hot spots are most likely to form – around combustion chambers, exhaust valves, spark plugs and the tops of the cylinders. Right away you can see that we have a problem with the way water travels through most stock engines. Typically the water pump forces all the coolant into the front of the engine block. Then, in theory at any rate, the water courses towards the back of the block, taking up heat as it goes. On reaching the rear of the block, the water moves up into the head through the large water passages interconnecting the cylinder block and the head. [illegible] through the thermostat or restrictor and enters the top of the radiator to be cooled.

However, the theory falls down because to prevent steam pockets around the tops [illegible] that would otherwise be trapped in the top of the block, they also allow the free passage of coolant from the block into the head. Also, as the spark plug bosses are often very close to the surface of the head, it is usual to find a series of holes in the block that direct water flow from the block up around the spark plugs. It should be obvious that all this coolant flow straight from the block to the head reduces the volume and slows down the flow velocity of coolant past the far-end cylinders and combustion chambers. The result is that the rear cylinders and combustion chambers run too hot, while those at the front run too cool. Thus to keep the back cylinders out of detonation we have to run the engine with a less than ideal compression ratio and reduced spark advance. All this costs us hp and engine efficiency.

Of course if we had the time and resources to experiment we could run each cylinder at a different compression ratio, higher at the front and progressively lower towards the back cylinder. If the race regulations permitted sequential fuel injection and ignition trim control we could trim the quantity of fuel injected into each cylinder, [illegible]g the hot cylinders richer and the cool ones leaner, and likewise with the ignition

we could increase advance on the cold cylinders and decrease it on the hot ones. All this effort would unleash additional hp and improve engine reliability, but the rewards are [illegible]

To give you some idea of how much 'free hp' is available in a race engine when [illegible] heads with improved air flow and better combustion chambers, which have allowed us [illegible] have picked up 8–10hp because we can now run a slower water pump speed, and even with less pump speed water temperatures have come down from 90°C and more to about 75°C. This has allowed a compression boost from around 12.2:1 to 14:1, which has lifted torque and hp at all rpms, and given around 13–16hp at peak. In turn more compression has permitted more aggressive cam timing, while maintaining the previous torque on 12.2:1 compression, for a 15–18hp gain at the top end. More uniform cooling and the elimination of steam pockets has allowed full power lean mixtures and increased spark advance for another 5–7hp.

Basically what we have done is to equalise engine temperatures as much as possible from front to back, and top to bottom. This has meant rethinking how much water we direct into the block and heads, and taking care of specific danger areas that expose the engine to head gasket failure.

It should be obvious that as the combustion process takes place in the combustion chambers, which are located in the head, it is the head where we have to focus the cooling effort. Yes, the block does require some cooling, but because by comparison the cylinders have such a huge surface area to give up their heat to the coolant, cooling difficulties in this area are minor. Consequently in a race engine we want to move a lot more water through the head, to take excess heat away from that area. However, if we follow the conventional coolant flow path this also means circulating more water through the block, but as the cylinders are already overcooled this would not be a good idea.

The solution is to change the coolant flow path. Rather than all the water flow from the water pump being directed into the block, then up into the head, why not take the [illegible] losses because we do not have to push bigger and bigger volumes of water through the engine, and frictional losses reduce as we allow cylinder wall temperatures to rise. Thus in a road race engine we would usually direct 65–80% of pump volume straight into the [illegible] front of the block, while the other half goes directly into the back of the heads.

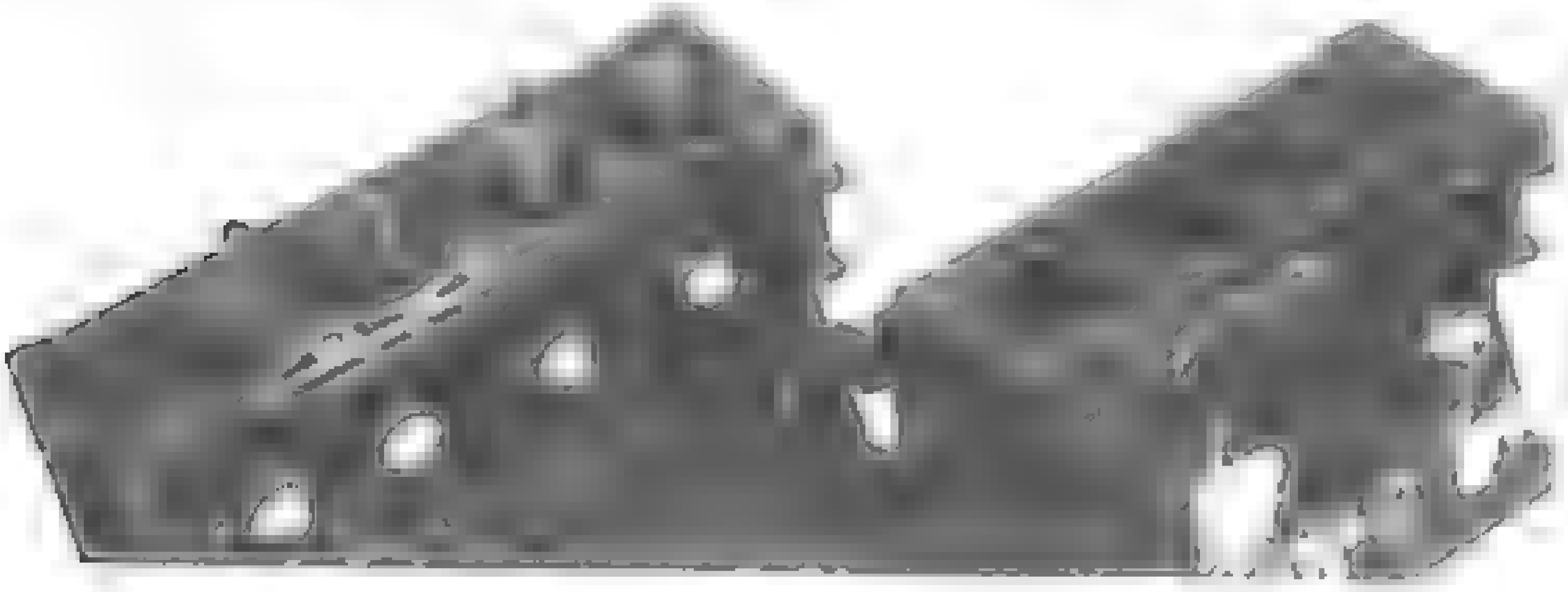
In a distance road race Chev small-block, 35% is taken into the block and 65% straight to the heads. However, because of some cooling problems in localised areas there is a further splitting of flow to direct water to specific areas. For example, the centre exhaust ports are side by side, with the result that the head overheats in that area, which in turn reduces head gasket integrity. The solution is to tap off a fraction of

the water being piped to the back of the heads and send it down a tiny 2.3mm hole drilled in the centre of the head below the exhaust ports. This reduces the head deck temperature and improves head gasket reliability. Down in the block, water flow pretty much goes to sleep around No 7 cylinder, and with reduced flow through the block that cylinder can begin to detonate. The solution is to run an external water line from the pump directly into the block at No 7. This provides a constant flow of cool water to mix with the stagnant, overheated fluid surrounding that cylinder.

One problem area with V6 and V8 engines is the manner in which coolant from the right and left heads comes together in the inlet manifold water passage before exiting through the thermostat or restrictor. Because flow is not the same in both banks – typically there is a 2–3% flow difference – water from the head with more flow will tend to reverse flow into the other head. This not only restricts coolant flow out of that head and causes it to overheat, but because there are two columns of water crashing head on into each other in the inlet manifold passage, there is a lot of flow turbulence. A few hp can therefore be wasted driving the water pump harder to force sufficient water volume through the engine to keep it cool. In a race engine the solution to this is to forget about bringing the water out through the inlet manifold, but instead to put a water outlet in the front of each head, if possible positioning it off-centre to favour flow along the exhaust side of the heads. Then coolant flow from the two heads can be brought together without turbulence through a pipe fabricated in the same style as two branches of an exhaust header blending together at the collector. The coolant flow restrictor or thermostat must be located in the system after the two pipes join.

Regardless of whether we follow the traditional pattern of cooling, with all of the coolant going into the block, then up into the head, or whether we use a split system with coolant diverging at the pump to both the block and head, we must concern ourselves with the size and number of coolant holes that interconnect the block and head. Obviously it is only at the rear of the block where we want volume flow of coolant going into the head. All other holes must only be as large as necessary to vent air and steam from under the deck of the block into the head, or to allow metered coolant flow from the block to a specific area in the head. Thus we may have flow

Take a few moments to study both blocks. The one on the left is the stock road car block while the other shows what BMW did to make it suitable for 1,000hp during the Formula 1 turbo era. The main change is the addition of coolant passages at the ends of the block and the elimination of six small steam holes between cylinders.



from the block to purge gas bubbles from certain areas such as from around the bosses

in that region by 70°C!

transfer in this area from block to head to purge gas bubbles, cool spark plugs, etc

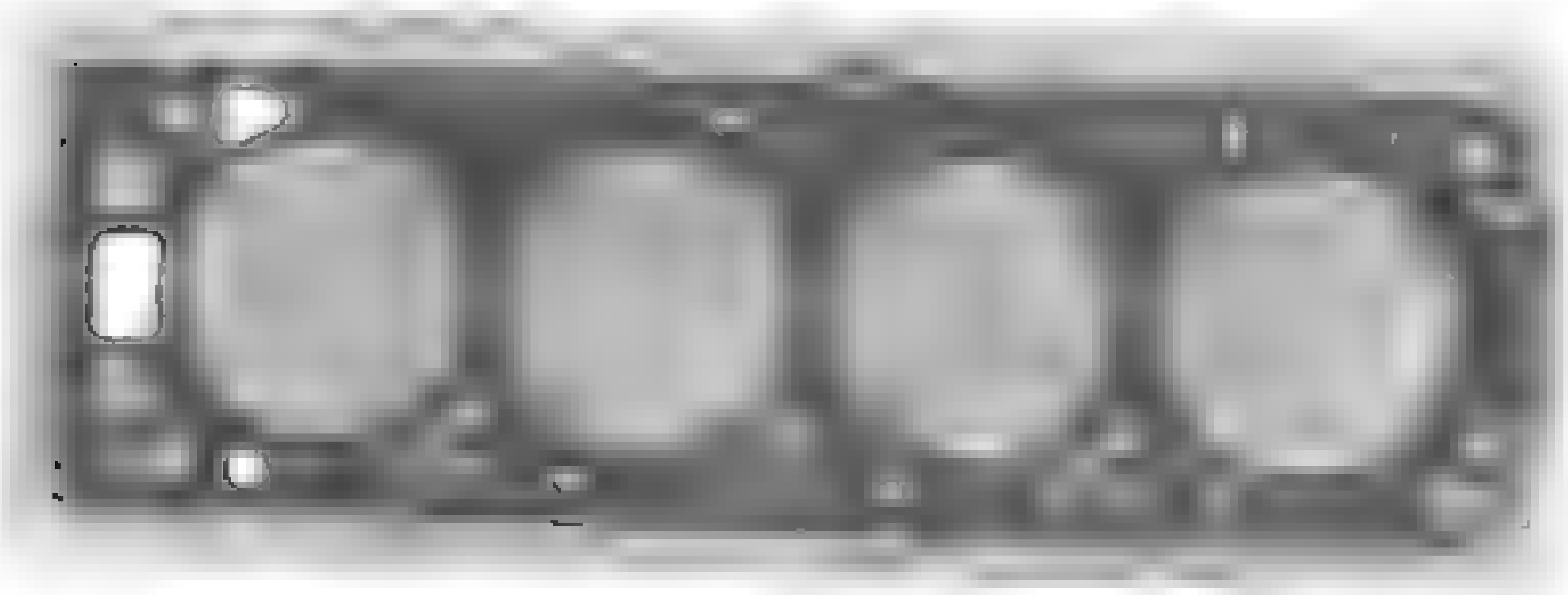
HEAD GASKET COOLANT HOLES

a competition engine, so the transfer metering holes can be smaller

It is at this point that we come up against a problem in that manufacturers of

The six small white holes allow only limited water flow from the block into the head

rear of the block



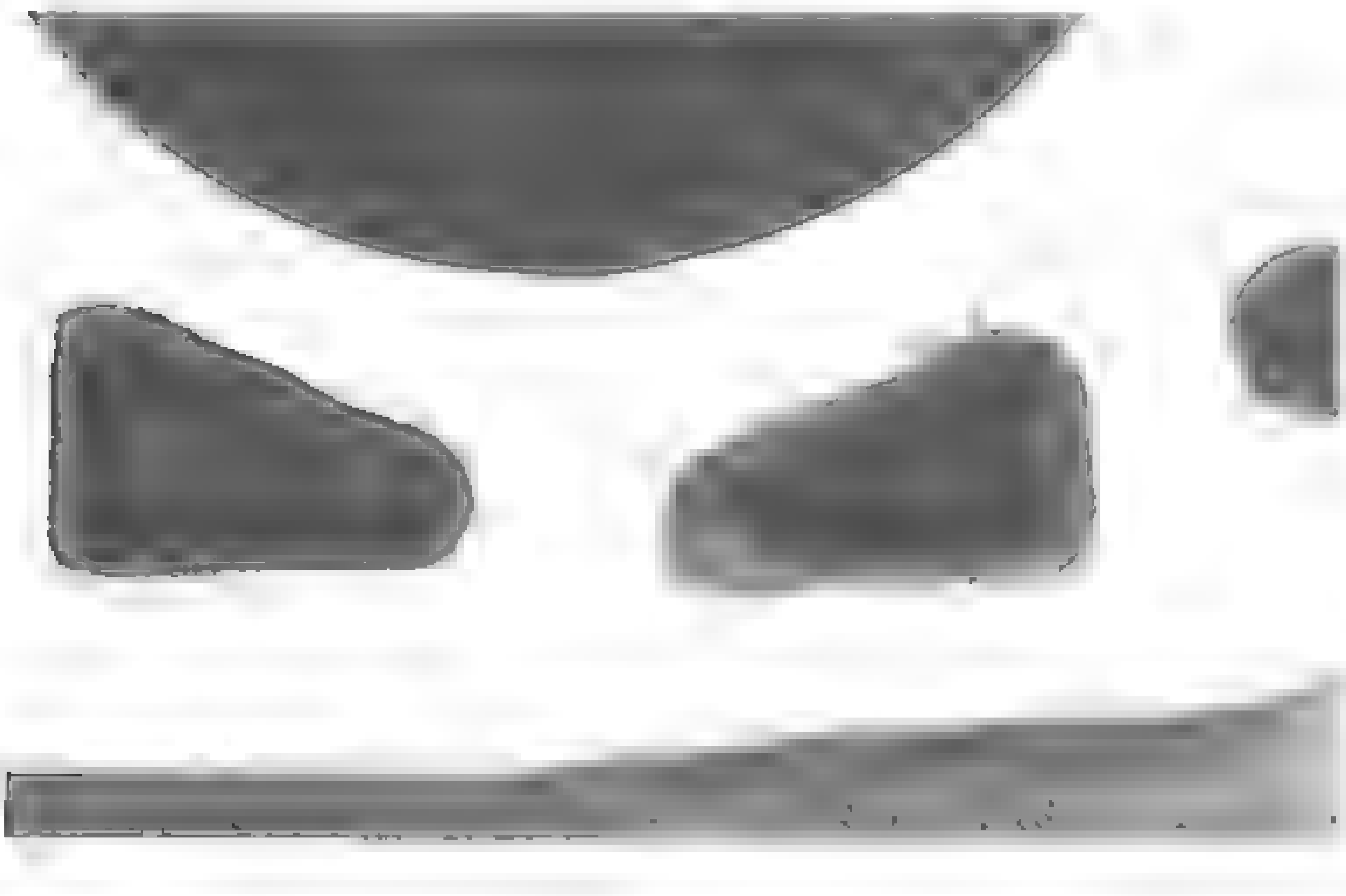
in soft aluminium pipe plugs. If the holes are smaller, gently knock in a short length of aluminium rod; obviously these plugs must not protrude above the block deck. After fitting them, drill the appropriate-size metering holes, which will usually be in the range of 0.060 to 0.125in. Note that in competition engines using a restrictor in place of a thermostat, the bypass hole is left completely blocked.

Where the transfer holes are an irregular shape they can be cleaned with a file or die grinder, then plugged with a silicone sealant such as Silastic. Coolant metering is taken care of by a length of aluminium rod, pre-drilled to the required size, being inserted into the silicone sealant. I prefer to run a drill through the sealant after it has cured, then force in the drilled rod, but some like to hold the pre-drilled rod in place then pump sealant in around it to plug the core hole.

Clearly these small metering holes in the block must match the holes in the head gasket to allow the venting of air and the flow of coolant into the head. Some tuners just punch large holes in the head gasket, but this is a job that will have to be repeated every time the head gasket is replaced. What is preferable if solid plugs have been inserted into the block deck is to use the head gasket as a template and drill each plug to match with the corresponding hole in the gasket. Naturally where the deck has been plugged with sealant and a pre-drilled rod has been inserted, it may be necessary to punch a matching hole in the head gasket.

Because we want the bulk of coolant circulation from the block to the head to take place through the large holes at the rear of the engine, we do not want them restricted in any way. With this in mind it is a good idea to smooth them out and match them up using a die grinder or a file. In a conventional flow system this encourages

The coolant hole on the right is badly blocked by casting slag. This must be ground out to encourage coolant flow through the back of the block and up over the end combustion chamber and exhaust port.



coolant flow right to the back of the engine, rather than flow short cutting through transfer holes towards the front. With a split-flow system a good flow of coolant in this region is also important; first, to cool the top of the rear cylinder adequately, and second, so that the majority of the hot water from the block goes into the head at this [illegible] from the water pump. This serves to balance out engine temperature front to rear.

COOLING HIGH OUTPUT ENGINES

While the modifications we have just discussed will provide adequate cooling in the [illegible] that single cylinder the coolant is returned to the radiator.

How this is achieved will vary from engine to engine, so I will describe the basic [illegible] arrangement is to introduce all coolant flow from the water pump into one side of the block. When the cylinders are siamesed we would have only a single inlet in the centre of the block. However, as I will explain later, if the cylinders are not joined and we are unable to seal the water passages between them we will require two or three inlets respectively for 4-cylinder, and 5- or 6-cylinder engines. The main thing we want to do is get maximum water movement up close to the top of the block and we want to equalise cylinder to cylinder temperatures as much as possible. Generally we would knock out existing welch plugs and tap the holes to accept suitable water fittings. Obviously because we have got rid of the stock water pump we have to cover that hole [illegible] the deck at the other end of the block.

[illegible] end, after having flowed over every combustion chamber. No, what we want in a high output race engine is for the bulk of the coolant to be divided up and enter the head along one side then flow across the head, exiting at several points into a water manifold on the opposite side.

Some race engines actually split water flow with about 70–80% going into the head and 20–30% going into the block. Using this more complex arrangement, which is really only necessary when we exceed about 250hp per litre, a water manifold is attached to both sides of the head. An obvious advantage of such an arrangement is that if in a 4-cylinder engine we have a water manifold with four outlets, the size of those outlets can be adjusted to equalise the temperature of each combustion chamber. Additionally the individual ports of the water manifold do not have to terminate where they attach to the side of the cylinder head. Rather what we can do is extend each port right into the water jacket such that the coolant is blasted into the main hot spot, [illegible] combustion chamber, then exit the head into the exhaust water manifold.

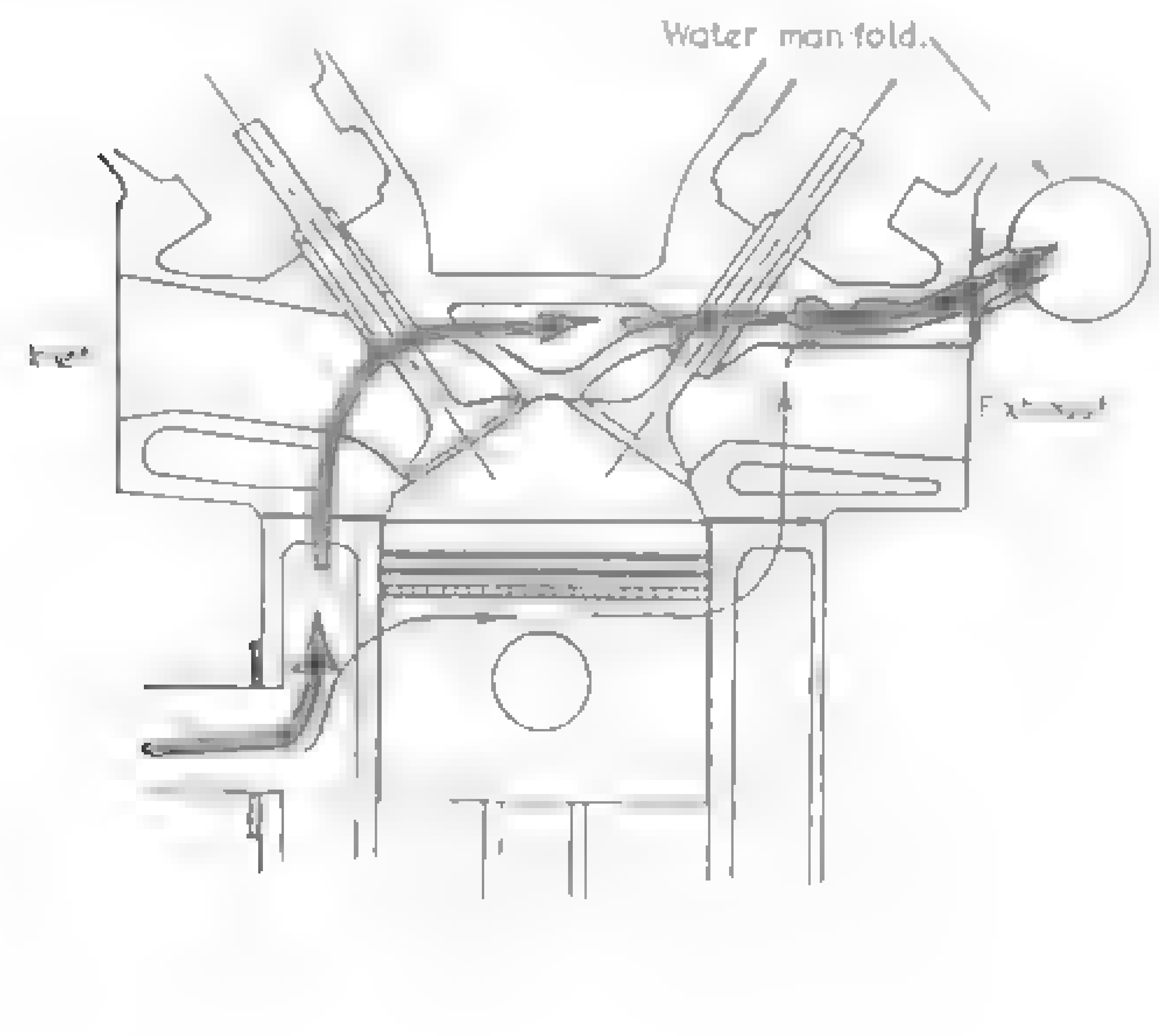
AN ALTERNATIVE COOLING METHOD

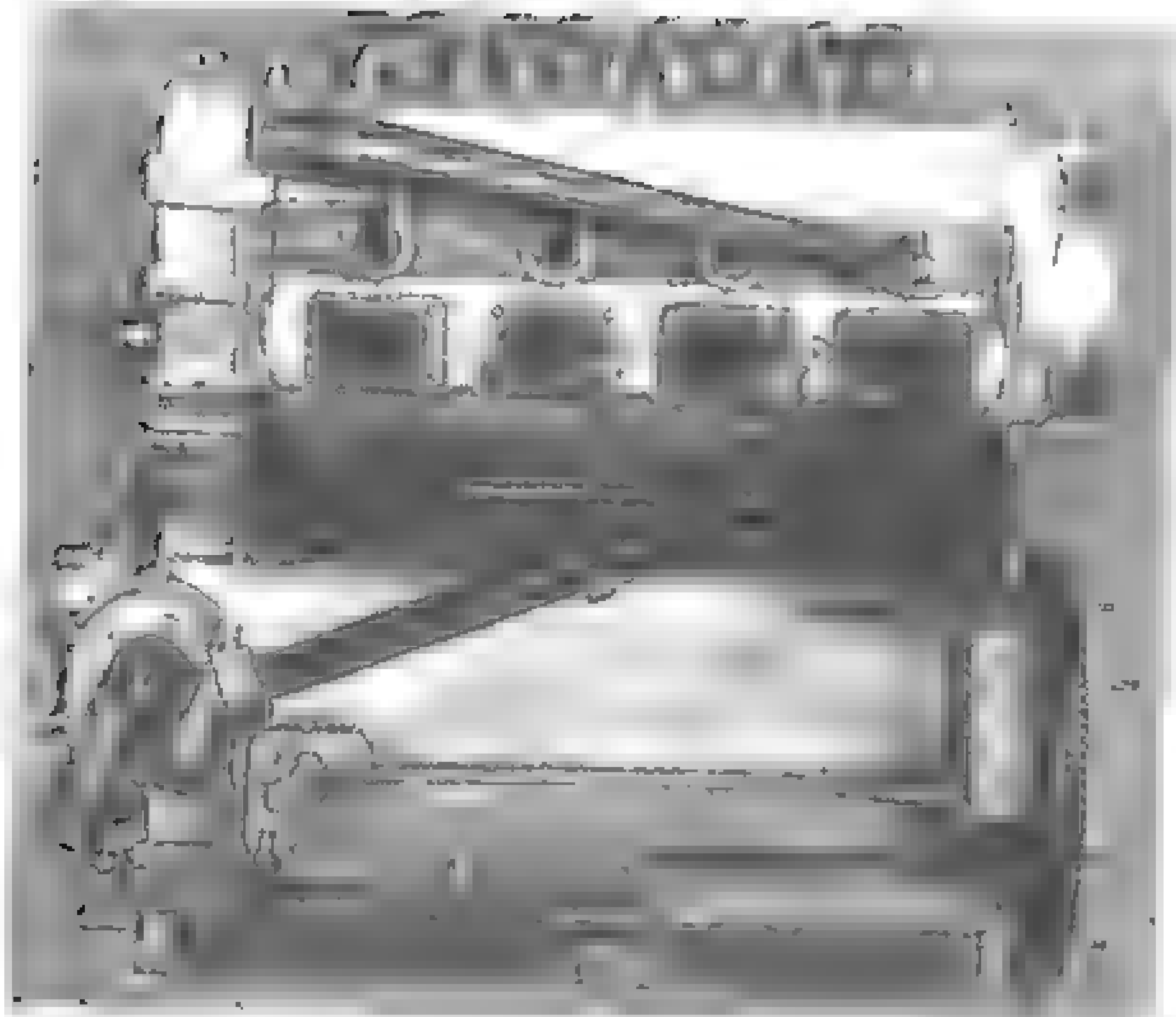
When we opt for the simpler arrangement of flowing all water into the block and from there into the head we are not able to exercise such precise control of flow across each combustion chamber. Of course if we have an engine management system offering individual cylinder fuel and spark trim, small temperature differences between combustion chambers can be compensated for, but in an endurance race engine where fuel consumption is a concern, injecting additional fuel into a 'hot' cylinder isn't a satisfactory solution.

What we have to decide early on is on what side of the head do we want the water to exit? If we decide it should exit on the exhaust side then most flow, around 65–80%, should travel from the block into the head along the inlet side. This will ensure that for the most part, water will move from the block, straight up into the head, turn 90° and flow over the combustion chambers, then past the exhaust ports whereon it enters the water manifold and returns to the radiator (Figure 13.2).

To achieve that flow path we may not have to do much more than punch bigger holes in the head gasket along the inlet side. In all likelihood, coolant holes in the block and head will already be larger than necessary to achieve the flow rates required and we will have to balance flow across each combustion chamber by adjusting coolant hole sizes along the inlet side of the head gasket. This may sound simple enough but usually involves long dyno sessions with temperature probes stuck down onto the exhaust side of all combustion chambers, testing head gaskets with various

Figure 13.2 Revised coolant flow path for a high output engine





The Waggott TC-4V 2-litre produced 270hp. That was almost 40 years ago and much of what I learned early on about high output engines was as a result of my being [illegible] *water manifold above the exhaust ports ensured that the exhaust valves and seats were* [illegible] *side of the head rather than use the path I prefer as illustrated in Figure 13.2*

combinations of different hole sizes to even out the temperatures. We can do the same sort of thing with track testing, but that is even more tedious.

Some tuners choose to always, if possible, have the coolant exit on the exhaust side of the head. It is frequently more convenient to locate the water manifold on that [illegible] maximise charge density, and flowing coolant from the exhaust side, across the combustion chambers, and out past the inlet ports gets the inlet hotter. While there is [illegible] engine and its hp level, be a more pressing consideration. Another view is that it is at times possible to achieve more control on the direction and quality of flow around the exhaust valve seats by having the water enter the head on the exhaust side; then, because the exhaust valves will be cooler, they will add less heat to the inlet charge. Also being cooler they are less likely to cause detonation after combustion has commenced.

However regardless of whether we settle on the coolant exiting on the exhaust or the inlet side, that decision will have a bearing on what side of the block the coolant should enter. The arrangement illustrated in Figure 13.2 with the inlet and outlet on the opposite side is correct, but we could still run into problems if the cylinders were non siamesed. For example, if the engine had 4-cylinders and a single water inlet in the centre of the block there would be minimal water flow around the outside of the end cylinders – most flow would be through the three gaps between the cylinders. If we had two water inlets located at about quarter distance and three quarter distance, flow around the cylinders would be evened up, but there is a better way. Depending on where openings are located it is often possible to block the gaps between cylinders with silicone sealant. We don't want to convert the block to being siamesed. Rather leave the top of the gap open down to about 25% of engine stroke, and close the gap from there to the bottom of the cylinder. This forces flow around the end cylinders and the small gap helps cooling around the top of the bore. Distortion of the bore is reduced, ring seal is better and hp rises slightly.

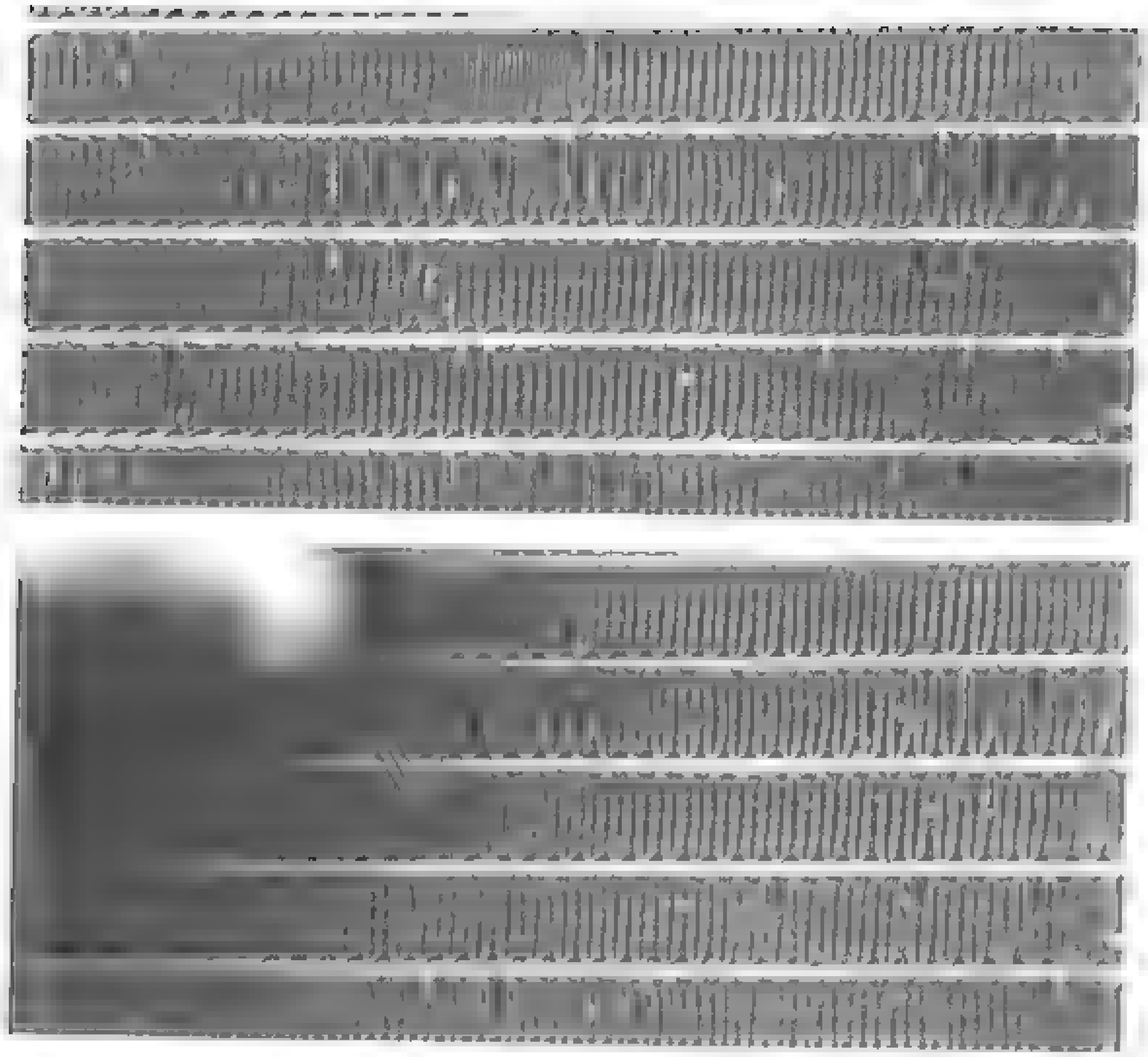
Usually it is possible to block those gaps, even if you have to add a plastic extension hose (use 4–6mm id) to your silicone gun. What though if you can't? Well, as pointed, out two inlets would be the way to go (three inlets for 5- and 6-cylinder blocks). If we were to put additional inlets in the opposite side we would simply compound cooling problems around the end cylinders.

With a block that has siamesed cylinders we can improve cooling and reduce bore distortion, if the siamesed walls are sufficiently thick, by adding a small coolant hole in between cylinders. This is precision work so the block must be set up on a mill. Obviously, if the hole goes off coarse the block becomes scrap! Basically the web joining the cylinders has to be around 6mm or more wider. If it is 6mm you will be restricted to a 2mm hole. The idea is to set the mill up so as to leave a minimum wall of 2mm either side of the hole. First you drill bigger access holes through one side of the block. Later these will be tapped and plugged with small pipe plugs. The coolant hole will be located as high as possible up under the deck of the block. The idea of this is to get rid of any steam that may become trapped there as well as move a little coolant through that area which is the hottest part of the cylinder.

RADIATOR MAINTENANCE AND DESIGN

The actual heat exchange between the coolant and air takes place at the radiator. Bugs and debris that restrict air flow through the core should be cleaned out to maintain cooling efficiency. Cleaning out balls of rubber thrown up off the track is always a problem. The easiest method seems to be to soak the radiator, after plugging all openings, for a day or two in a tray of solvent. Then direct compressed air at the back of the radiator. Any balls that will not budge can be carefully picked out. A large tray of solvent is both a health and fire hazard so be very careful. Any fins which are bent must be straightened, as bent fins present as much impediment to air flow as trapped rubbish. The radiating efficiency will be improved further if the radiator is regularly repainted with matt black paint. This will also serve to increase the life of the core, by reducing the effects of external corrosion.

However, when it comes to a competition radiator many more factors are involved, such as whether it will be constructed of copper or aluminium, how many rows of coolant tubes it will have, what fin count will be most suitable, etc.



[illegible] ngth. Copper is [illegible]

[illegible] avoid coolant tubes [illegible] re, the tubes must

[illegible] row. Aluminum, on the [illegible] stronger, so the tubes can

[illegible] such an advantage, but [illegible]

[illegible] es. The water [illegible]

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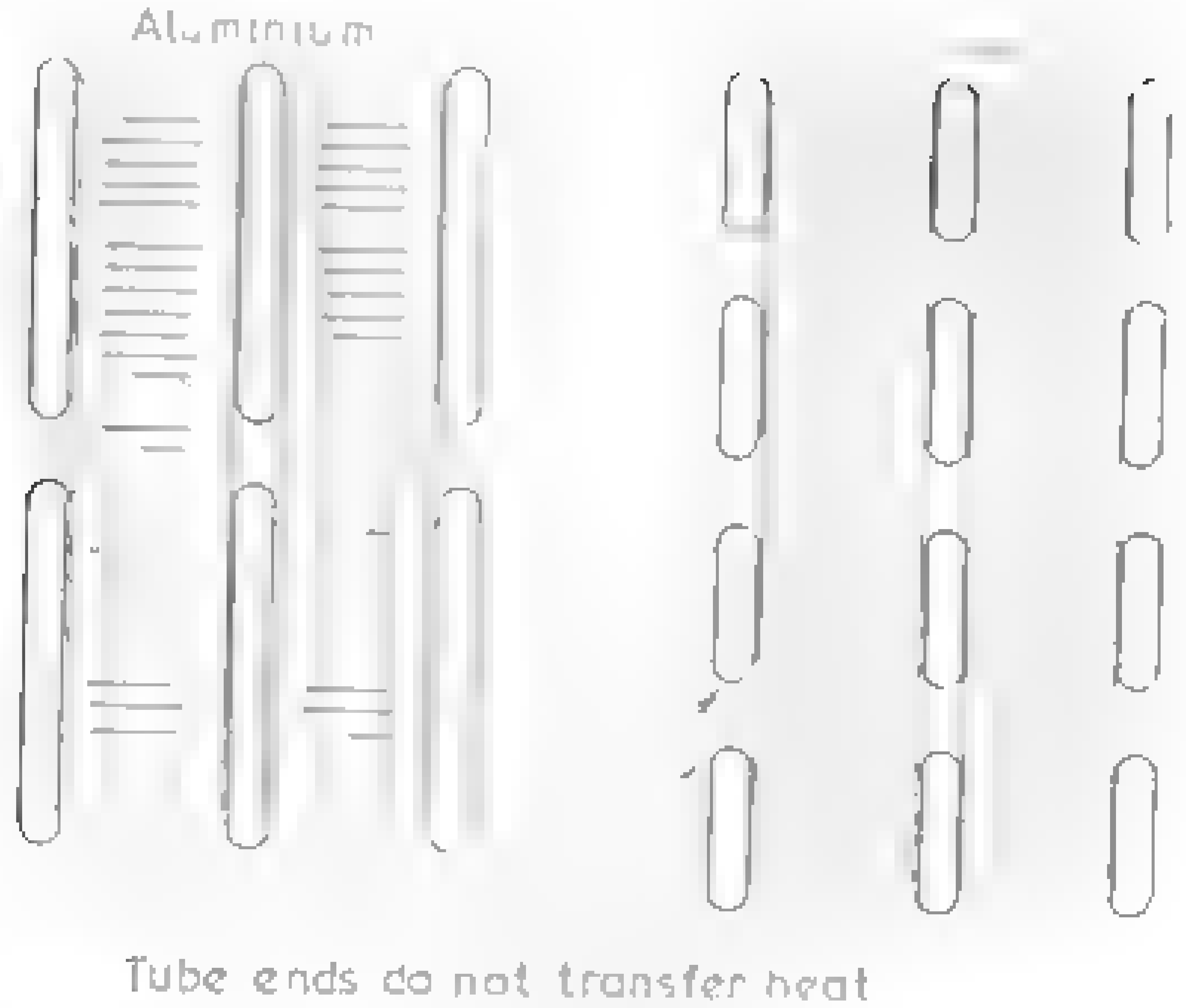


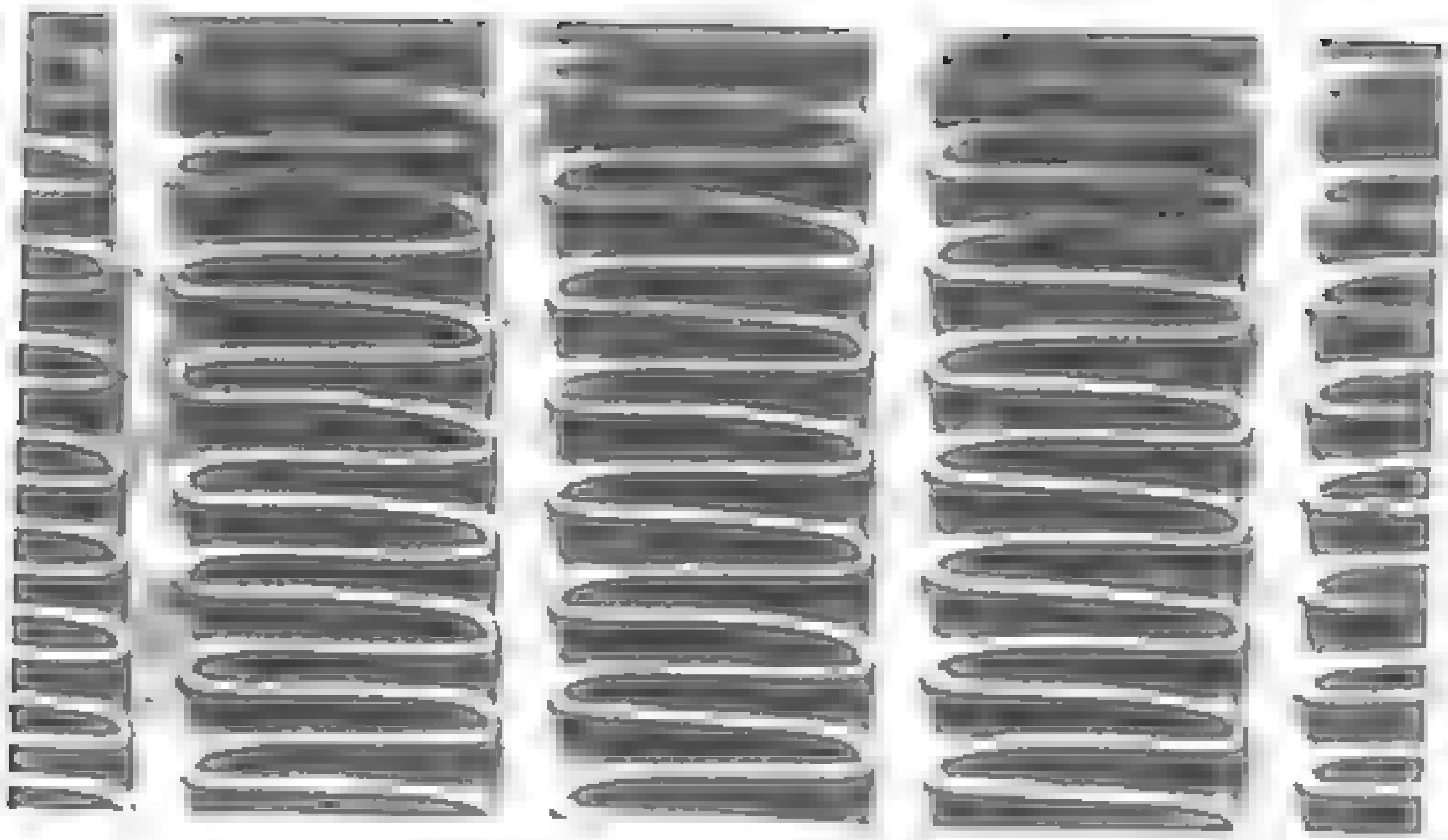
Figure 13.3 Aluminium radiator tubes are twice as wide as copper tubes

of the tubes have no finning, so there is no heat transfer of any significance from this area. Th[illegible] more heat to the fins.

Another big difference between the radiators installed in road cars and race cars is [illegible] regard to their construction [illegible] a funnel. The rate of flow is controlled by the size of the hole in the bottom of the funnel, the size of the hole at the top and the amount you pour into the funnel, even to the point of overflowing, really does not change the rate of flow through the funnel by very much.

The only way around this is to construct a radiator with many more tubes – but not necessarily side by side, as this just adds to the air flow problems unless we can build the radiator wider or taller (depending on which direction the tubes run). What we usually need are more tubes placed one behind the other. The exceptions would be competition cars operating at speeds too low to force sufficient air through a thick radiator core.

Typically a road car will have a two-row or perhaps a three-row copper radiator, while a race car will perhaps have a radiator that is four or five rows deep, and to assist air flow the tubes may be spaced a little bit further apart across the front of the radiator. So, whereas a road radiator may have the tubes closely bunched at, say, 2·

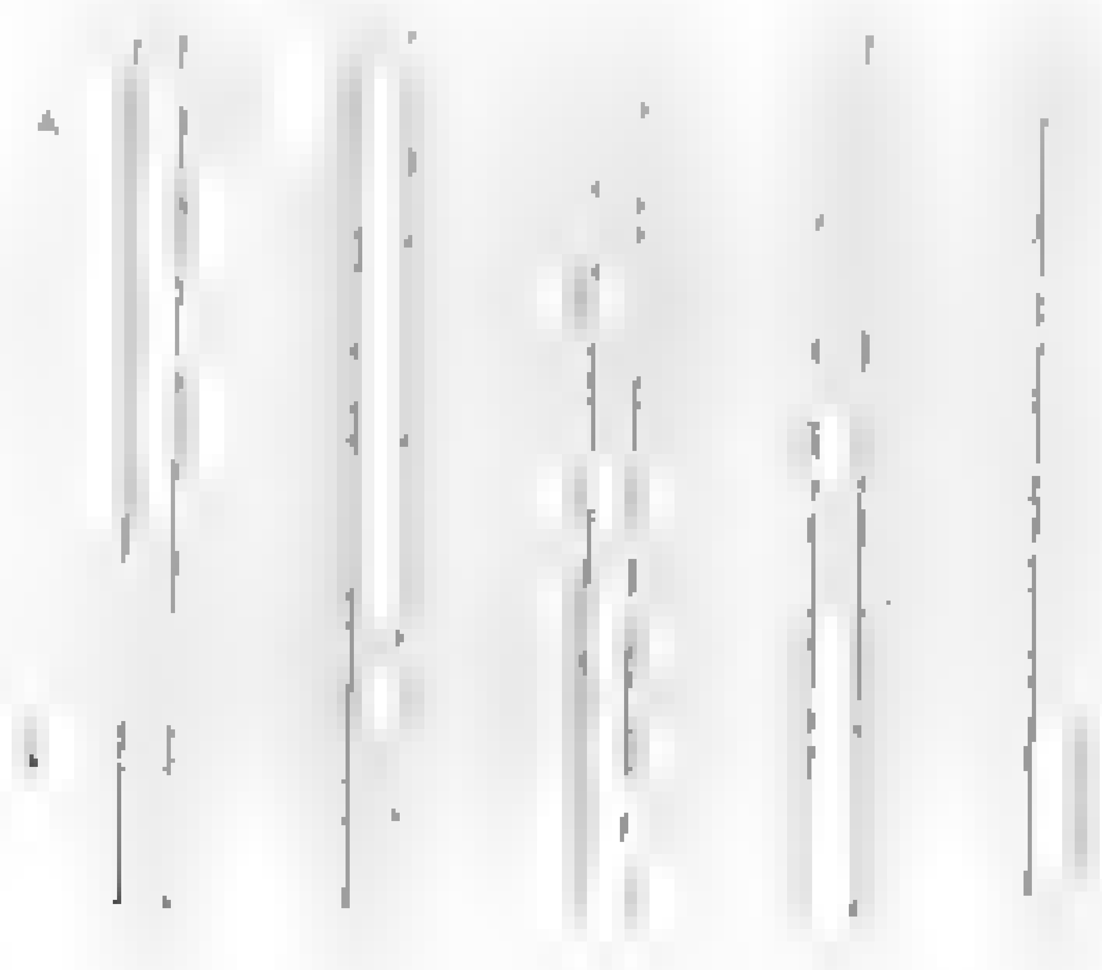


Louvres on the fins increase air turbulence across the surface of the fins, thus raising their heat shedding capacity. However, this also cuts air flow through the radiator, on t... so deeper radiators must have less fin densit...

tubes per foot with a fin count as tight as 17 per inch ... competition radiator may sp... out the tubes to 12 per foot with a fin count opened up to 10–12 per inch, or even as low as 8 fins per inch in dirt track cars where there is inadequate protection to prev... dirt getting in and blocking air flow.

If you have a close look at the fins you will better understand why a high fin count is so bad for good air flow through the radiator. Rather than having a smooth flat face, the fins are very finely louvred across their surface. These tiny louvres create turbulence to increase their radiating efficiency ... nd as they are packed more closely together it is this turbulence across their surface that cuts air flow through the radia...

Figure 13.4 Competition radiator constructio...



When it comes to correcting a cooling problem racers often go about it the wrong way; they adopt the road car radiator mentality of closely bunched tubes and high fin counts. When that fails to work they may even exacerbate the problem by moving to an even higher fin count! Because the overheating just gets worse they drive the water pump faster, which in turn increases parasitic losses at best, or else runs the pump into cavitation. Alternatively they may throw out the restrictor, so water pressure in the block and head goes down (remember that we want about 30psi here), so big steam pockets develop, the engine detonates and loses power at best, or perhaps lunches a head gasket or piston.

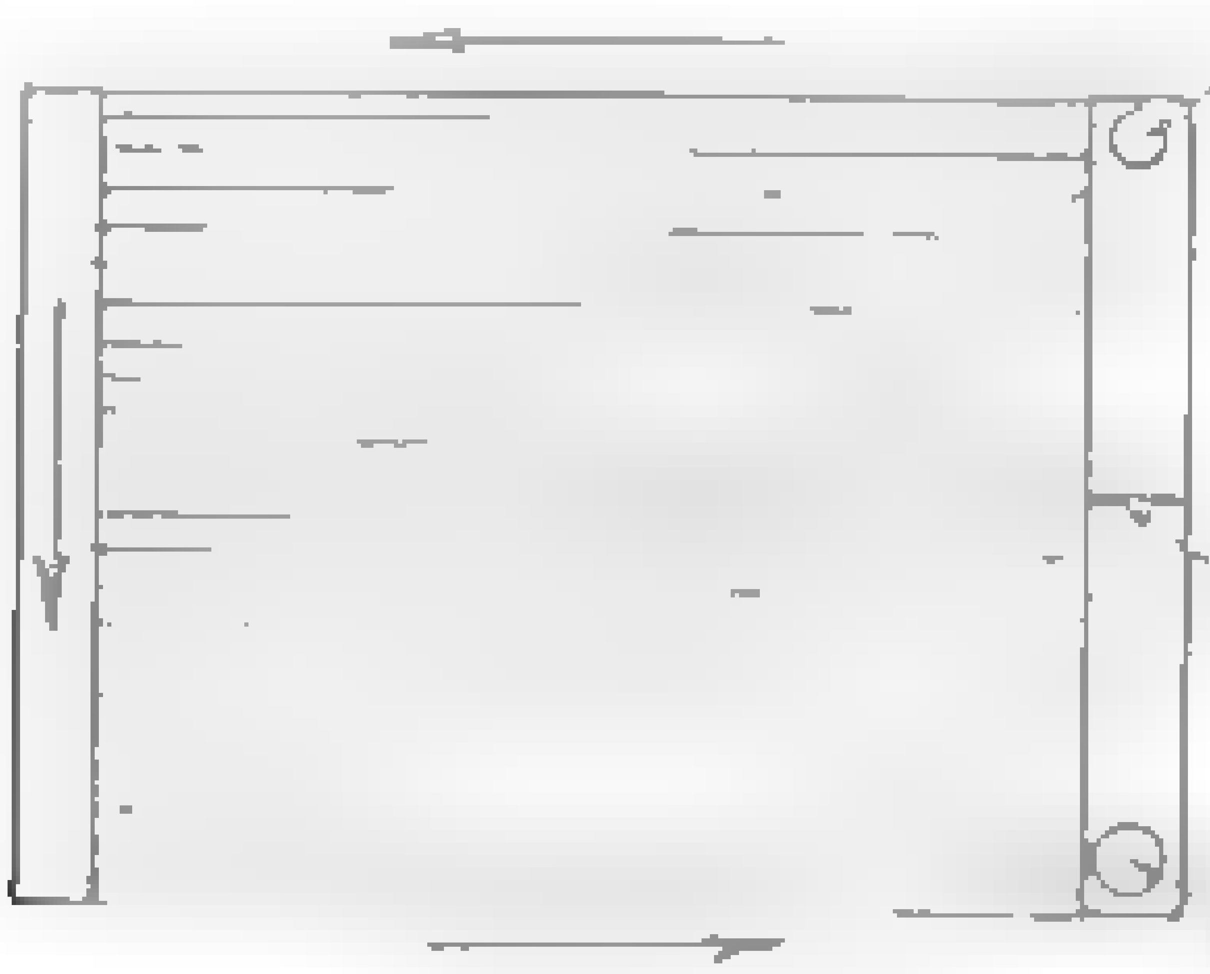
Really what we want is a large number of radiator tubes but a low fin count. This costs more money, but you may save a great deal in the long run. If the radiator brings the temperature down too far, run the pump slower, but fit a restrictor with a smaller orifice to get block pressure back up to 30psi. Less pump rpm will cut parasitic losses, and reduced air flow resistance through the radiator may see a speed gain on fast straights.

DOUBLE-PASS RADIATOR

Another way we can increase the heat dumping ability of the radiator is to build it as a double-pass unit (Figure 13.5). As you can see when a splitter plate is welded in one tank the effect is to force all the coolant through half the number of tubes, but over twice the distance as the water flows from one side to the other side across the top half of the radiator, and then back again through the lower half. Providing the water pump is up to the task a two pass radiator will dump 15–20% more heat than an otherwise identical conventional single-pass radiator.

This happens in spite of the flow rate through the tubes dropping 25–30% due to increased friction and water turbulence. What then accounts for the increase in efficiency? Obviously the water travels twice as far, giving up heat as it goes. However the real gain comes as a result of increased water turbulence in the tubes. At low flow rates the water clinging to the tube walls pretty much stalls, and in effect becomes an insulator which reduces how much heat is shed by the main body of

Figure 13.5 A double pass radiator halves the number of tubes and doubles their length improving radiator efficiency



coolant flowing through the centre of the tube. When the flow rate is pushed up by effectively halving the number of tubes, water turbulence increases and this has the effect of breaking up and scrubbing away this stagnant insulating blanket clinging to the tube walls. This improves the heat transfer rate from the coolant to the radiator.

REGULATING COOLANT TEMPERATURE USING A THERMOSTAT

A high temperature thermostat, which maintains the water temperature at 88–90°C, is usually fitted to production cars, but that is not the temperature for best power. The 88°C thermostat is fine if you want good heater efficiency in winter, and the higher temperature partly compensates for the stock manifold and carburettor being unable to vaporise the fuel properly, but apart from that there is no reason for its use.

Some feel that high engine coolant temperature in the range of 90–100°C is good for power. The theory is that anything lower just takes away heat energy, which we should be using to push the pistons down and produce more hp. To an extent this is true at low engine speeds and small throttle openings, but generally in a full throttle, high-rpm dyno test an engine will make more power at coolant temperatures in the 70–80°C range. Only dyno testing will confirm what is ideal for a particular engine, but in my experience I rarely see an engine that makes more power running higher than 84°C or less than 68°C. I have seen engines gain 2–3% hp when the temperature was dropped from 90 to 70°, conversely I have seen gains when the temperature was pushed up from 65 to 80°C.

The water temperature in a road car can be lowered by fitting a cooler thermostat, providing that the radiator is large enough. Whenever a thermostat is replaced, use the correct type. Many modern engines use a dual function bypass-type thermostat. When open to allow water flow out of the head into the radiator, this type of thermostat also closes off the engine bypass port. If an ordinary thermostat is fitted this port will be open, thus allowing a large volume of very hot water to circulate in a closed loop within the engine. This may lead to the engine overheating because of the volume of hot water from the bypass mixing with coolant from the radiator and significantly raising its temperature as it flows into the block.

A bypass type thermostat improves cooling system efficiency by blocking off the bypass port when the thermostat opens. Never replace this type with a conventional thermostat.

RADIATOR COOLING FAN

The fan fitted to most cars consumes a good deal of power, which could otherwise be [illegible]

Therefore a fan will not be required in competition vehicles. I have never had one [illegible]

Liquid-cooled vehicles with rear-mounted radiators (eg Imp and Renault) or [illegible] certainly require a larger cooling fan and also a separate expansion tank connected to the radiator. Some engines have a tendency to overheat if the radiator header tank is not completely full to give proper water flow down through the radiator core; an expansion tank ensures that the top tank is always full.

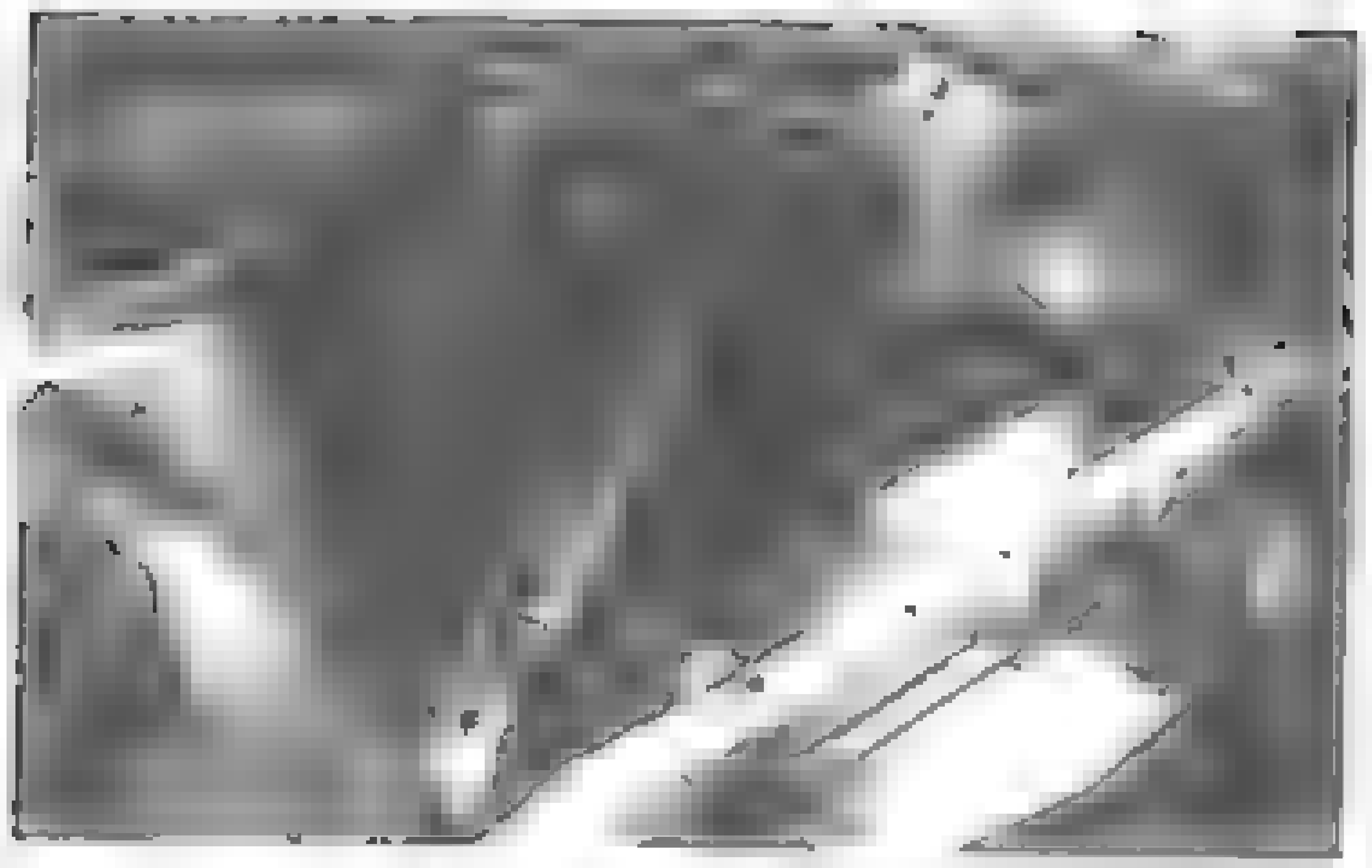
If a front mounted radiator is installed, the fan can be removed and you can be sure of maintaining the correct coolant temperature under competition conditions. This is a simple modification on the Mini, particularly if of the Clubman body design.

Whenever an electric fan is fitted it should be mounted behind the radiator to [illegible] current draw of around 10 amps, so be sure that the alternator can handle the [illegible] pump slowed and fuel pressure at the injectors dropped.

Some drivers run the electric fan all the time to get sufficient cooling, but this [illegible] aerodynamics. Others use a thermostat control to switch on the fan at a pre-set temperature. Personally I feel that this just complicates things. I prefer to see the driver [illegible] fan on into the pits', etc, he will not have to worry about studying the temperature gauge in such situations and make a decision as to when the fan should go on.

HELPING FLOW THROUGH THE RADIATOR

Another situation where an electric fan will provide an improvement in cooling is when air flow through the radiator is poor due to excessive pressure build up in the engine bay. Really we need to work to exhaust air out of the engine bay to get air



Building a duct around the radiator helps more air to flow through the core, improving cooling

pressure down as low as possible behind the radiator. However, while a solution is being found an electric fan may provide a short-term fix. Reducing air flow under the front of the car will help, as will opening holes into the wheel tub area. At times the problem can be traced back to an inappropriate bonnet air scoop, which may be providing good air flow into the engine air intake, or cooling air flow over the turbocharger, but which is 'over stuffing' the engine bay with air.

Of course every effort must be made to ensure that all the air entering at the front of the car actually passes through the radiator; we do not want any air bypassing it by going over, under or around it. This means blocking off any openings around the radiator or else constructing a duct from the front of the car that seals up against the radiator's edges.

THE BENEFITS

To some, no doubt, all of this discussion and concern over cooling may appear to be over the top. However let me remind those folk that during the Formula 1 turbo era the major problem confronting teams, even allowing for their vast resources both monetarily and technically, frequently related to engine cooling. While readers may not be pushing the boundaries to the same extent as the F1 crowd, the fact remains that when we generate very high hp levels out of relatively small mass produced stock based road engines we are moving a considerable distance away from what the manufacturer's powertrain engineers ever envisaged, or made provision for. Consequently if we are to attain such power outputs with relative reliability we have to spend much in the way of time and resources to take care of engine cooling.

The other aspect is, if we are ever going to attain big hp then to some extent that outcome will be attainable only when we equalise, as much as practicable, combustion chamber and cylinder temperatures. Until that is achieved there will always be at least one cylinder giving less than 100%. And if we don't have individual cylinder engine management capabilities the situation is even more depressing as the cylinder that is the detonat[illegible] will dictate just how much compression, fuel and spark we give all the other cylinders.

Chapter 14

Power Measurement and Tuning

M
themselves with fine tuning on an engine dynamometer. No matter how much we may try to fool ourselves, there is no way that an engine can make top performance tuned by ear and

with which I have not had a lot of experience, more often than not I gain something like 10%.

ROLLING ROAD DYNO

There are two basic types of dynamometers, the engine dyno and the rolling road or chassis dyno.

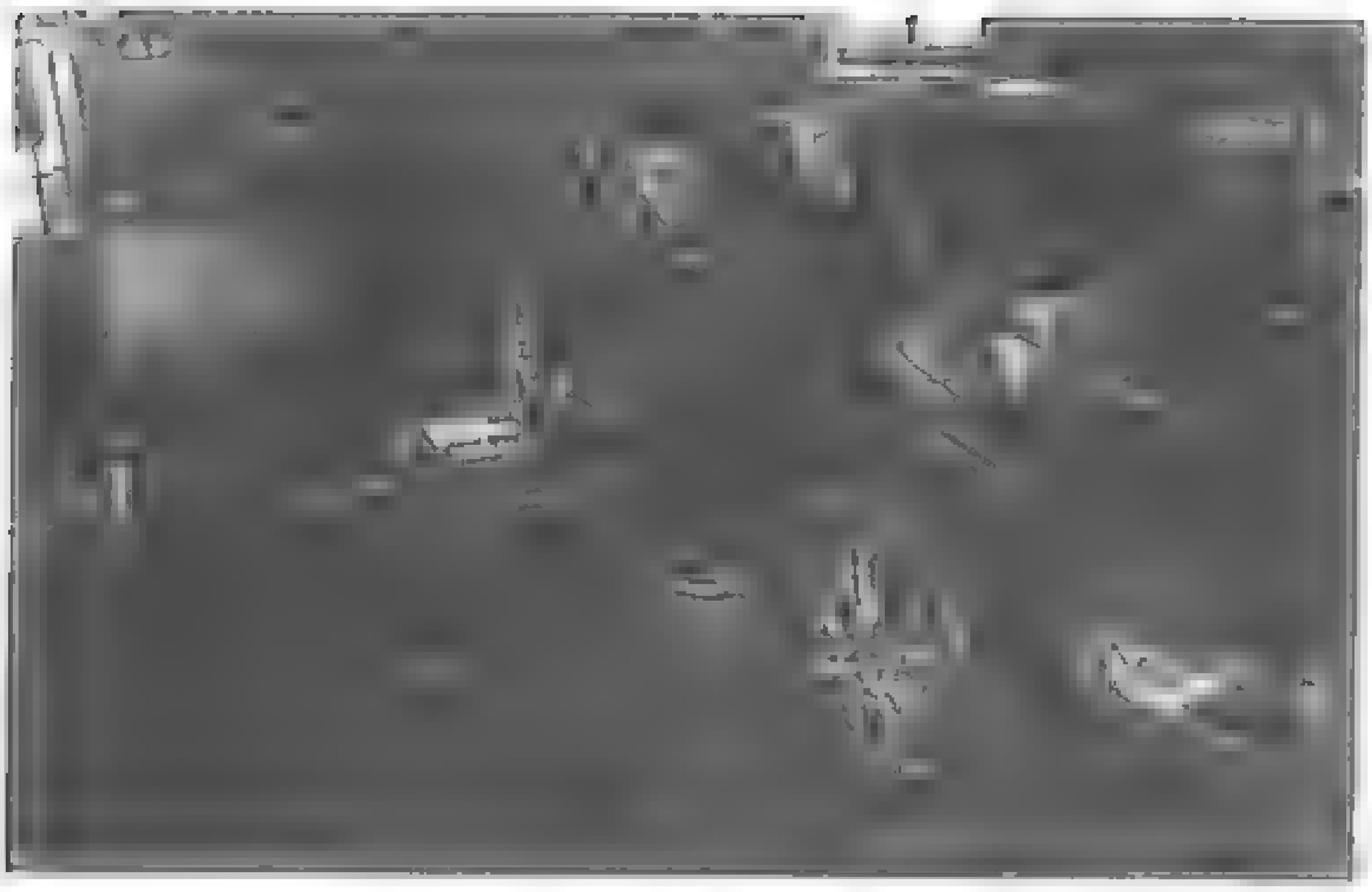
road. The dyno may be either an inertia type or a brake type and the latter can be further categorised as being either a water brake or an eddy current brake.

The inertia dyno employs a large diameter drum which is accelerated by the vehicle's driving wheels. A large diameter drum has more inertia than a smaller drum

manufacturer knows the inertia value so when the wheels accelerate the drum the torque can be computed. Accurately measuring this acceleration over a small time

power. For example, if a car fully developed for rally work usually produces around

On the other hand if it showed 135hp then you can be sure something is amiss.



Above: *Under the control of a comp... ...ddy current type ... road has much in its favour*

Right: *This is the eddy current brake. The lever on the right connects to the ... and the signal is taken ... the ... computer software to calc... engine output. What appears ...*

...much more useful rolling road is the brake type. This utilises some form of brake, or varying load, connected to the rollers that the wheels turn. Years ... type employing hydraulics to load the wheels. These are ... quite useful but an eddy current brake is much more user friendly. The ... unit looks like a large alternator. Applying electricity to ... the load to resist wheel rotation. The ... computed into torque, and then into hp. Because a constant load can be applied the eddy current rolling road is excellent for ... ignition and fuel injection.

The power outputs computed by inertia and brake rolling roads puts them ... significant odds with each other. While an inertia type like the D... something like 120hp, an eddy current road will most likely indicate close to 104hp, something like a 14 or 15% difference, and as the power incre...

THE ENGINE DYNO

The engine dyno is very sensitive and will give a clear indication of precisely what [illegible] recorded so that you will know exactly how to set up the engine to perform at its best [illegible] what to do to get the best low-end power when you need it. You will know what combination gives best overall power, and if you need maximum power you will know [illegible] leave anything to memory.

Now don't get me wrong — engine dynos are definitely not the 'be all and end all' of engine tuning. There are clearly definable limits to the usefulness of most engine dynos due to the static nature of the load applied to the engine. Only on computerised dynos like the Superflow SF-800 and its more recent siblings can you check how crisply an engine will accelerate, or what its throttle response will be like.

Usually the areas where you will get caught out are those of carburation and [illegible] carburettors that, on the dyno, give seemingly identical performances, but on the race circuit there will generally be one set-up that is superior to the rest, allowing improved lap times or maybe better performance in race traffic if you have to drive through the field, as in speedway racing.

LIMITS TO RELIABILITY OF HORSEPOWER FIGURES

There are several types of engine dynos in use, and while all do a good job in enabling a motor to be set up correctly, do not take a lot of notice of the power figures [illegible] accurate dynos. Instead, in many cases they make do with older or new, less [illegible] done on the same dyno. Otherwise you may try out some new trick part and find that it gives you 8% more power on a dyno on the other side of town, when in fact you had lost power with that trick. It was just that the dyno was reading higher than the one on which the motor was originally tuned (Table 14.1).

Another trap may also result from the use of these older dynos. Within the trade it is generally well known whose dyno gives the highest, most distorted power outputs. To sell an engine or even go-quicker bits, some have resorted to testing their motor or equipment on such a dyno to make it appear from the dyno test sheets that the motor is much hotter than it really is. Before you buy a motor or equipment on the pretext of it being more powerful, or more something, be sure that you know on which dyno it was tested and make enquiries as to the fidelity of its power readings. Also ensure that the engine number is quoted on the dyno sheet and check that it corresponds with the motor you are buying, otherwise you could be taken for a ride at considerable expense.

Table 14.1 Dynamometer comparison using 350cu in Chevy test engine

	Dyno A		Dyno B	
rpm	*hp*	*torque (lbf ft)*	*hp*	*torque (lbf ft)*
5,500	472	450.7	477.7	456.2
6,000	505.3	442.3	532.7	466.3
6,500	567.6	458.6	560.5	452.9
7,000	601.4	451.2	582	436.7
7,250	606.3	439.2	582.8	422.2
7,500	611.8	428.4	584.8	409.5
7,750	622	421.5	551.4	373.7

Note this motor was not altered in any way between tests. The hp and torque figures have been corrected to compensate for changes in air density.

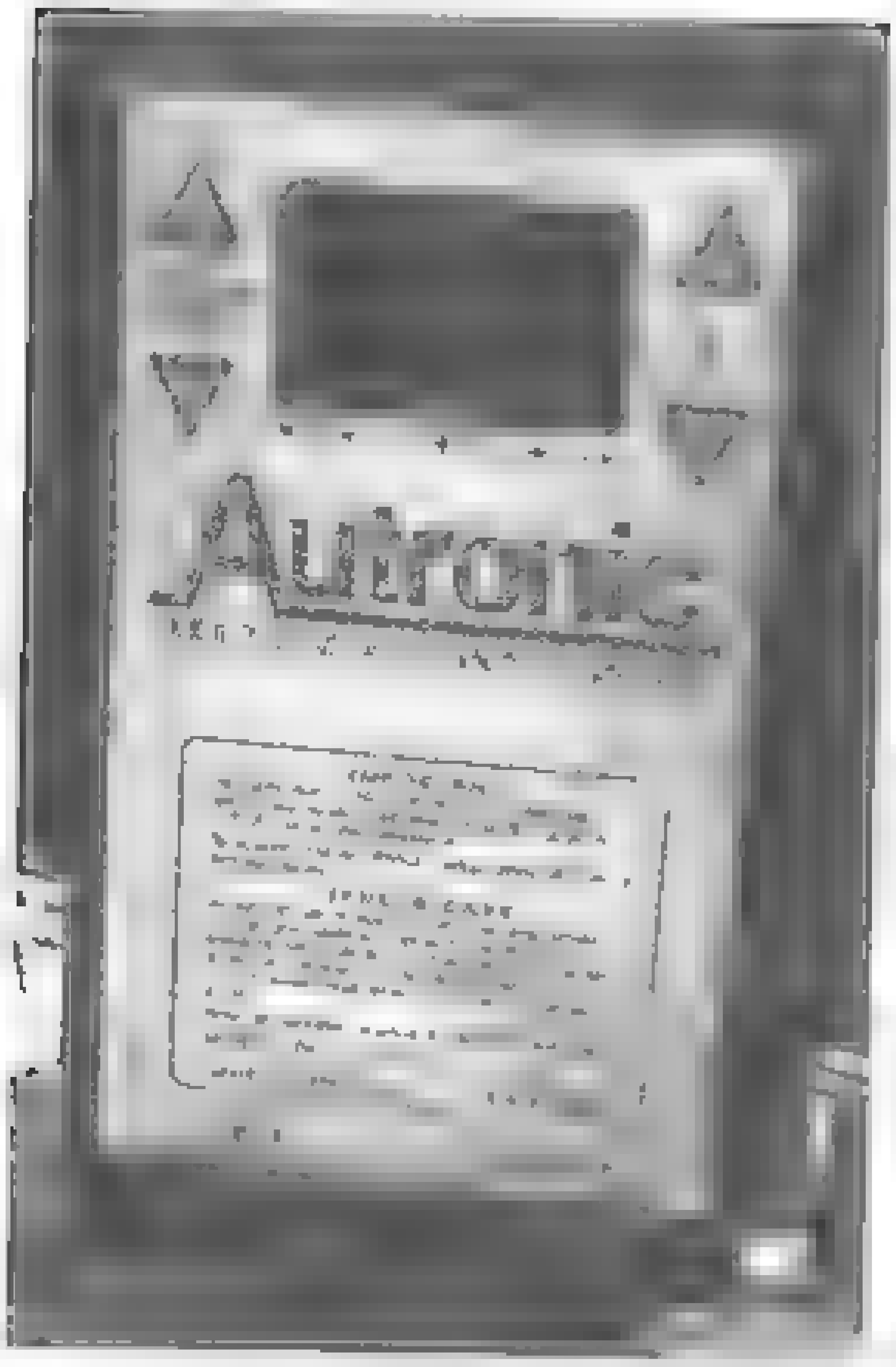
While we are on the subject, keep in mind that there is no guarantee that a company does top-quality work just because some big names use their engines or equipment. What you buy and what the top name buys (or is given) are not necessarily of the same standard. This is unfortunate, but it is a fact of life. Power figures get thrown around rather carelessly by some firms, I refer to them as 'paper horsepower'.

To cite an instance of this, I remember a top engine builder advertising that his particular engine produced in excess of 200hp. This was surprising as most comparative engines were producing around 15hp less. A couple of these trick engines were bought for evaluation and one produced 186hp and the other 193hp. Now these were guaranteed to produce a minimum of 200hp. About the same time a top-name driver ran two engines identical to these, and on the same dyno they did indeed produce in excess of 200hp. The two dud engines were stripped and the deck heights were found to be all over the place, combustion chamber volumes had not been equalised, the valve seats were not concentric or of equal width, and inlet ports were not of the same configuration. A pair of replica engines were built up using the same design features, but with everything done correctly, and these produced 208 and 209hp. That must prove something about doing things the correct way. It also proves that you do not get the best just because you pay for it. To get the best in many instances you will have to tidy up the loose ends.

Top firms have their problems with outside suppliers too, so it is not always entirely the fault of the engine builder when things go wrong. Obviously he must bear some responsibility if he has not made an adequate check to ensure that the outside work is up to standard. Just a few years back a Formula 1 engine constructor was having a series of crankshaft problems. When an investigation was made, it was found that the supplier had made a machining error during manufacture, due to an improperly dressed grinding wheel. A whole batch of cranks was affected and those that had not been used were scrapped. You can appreciate how much harder it is for a private individual to get top-quality work when a World Championship engine constructor cannot always get it.

PERFORMING A STANDARDISED TEST

Even if you are using the newest, most sophisticated engine dyno, the performance figures obtained will be pretty much meaningless, except for perhaps some sort of shallow bragging rights, unless you do all you can to perform a 'standardised test'.



Left: Tuning using an A/F ra... to tune the ... with little risk of engine ... ne too lean. However ... ere has not been con... e exercise may have been a waste of time and ...

This dirt track sprin... being tested with the same duct systems as run when instal... car, even going as far as having a ... of ... The induction atmosphere is dyno c... which the operator reasoned ... is OK ... there was a good flow of ... into the room, and anyway the methano... mechanical fuel injection system is not exactly a precision metering d... However, as the atmosphere changed over the several days that cam compari... being done, how was he ... exactly what cam was really ... suitabl...

By this I mean you have to carry out your testing in a controlled environment as much as possible so that the engine being tested today will produce an identical set of numbers tomorrow, next month, or next year. That may sound like a tall order but in [illegible] and 0.7–1.0% the norm, we have to do all possible to determine if we are making real headway with development.

There are three areas of concern. First, the atmosphere influences power so we need a standardised atmosphere in our dyno room. True, we can apply correction [illegible] being useful in ordinary testing such corrections are not a good idea for intensive [illegible] problems that can be easily rectified revolve around the source and availability of intake air for the engine. It is pointless testing with the engine dragging in hot air from within the dyno room, that's pretty obvious, but it is no better for it to be pulling in air from outside contaminated by exhaust gas, or by water spray from the air conditioning tower on the roof, or by solvents from the paint shop or parts washer room.

Equally important as the quality of air available for the engine is the volume of air available. Obviously we don't want the engine drawing a vacuum and gasping for air, equally we don't want the air supply pressurised either. Therefore the air supply system, whether it be unconditioned outside air or air from a conditioning unit, must supply at [illegible] this can be very easily overlooked if it is not your normal practice to measure the [illegible] pressure in the air supply ducting where it connects to the engine's inlet.

Operators of wheel dynos have many more problems in this area. The best thing you can do is have a good exhaust extractor fan drawing on an oversize flex pipe connected to the exhaust. If you find that your eyes water when testing cars running race fuel or methanol the extractor fan isn't doing its job, so the engine will be drawing in exhaust gas as well. Air supply quantity can be an issue too as noise [illegible] to blow air into the garage.

Second, engine variables must be eliminated. This means testing with the [illegible] uses an ignition system affected by battery voltage then the car's battery and charging system, together with any other high current draw components need to be connected. In the past we were content to hold water and oil temps within 5°C. That's still okay for ordinary development, but in tightly controlled classes where we are constantly [illegible] temp in the brake to ±3°C.

Also we have to test with a 'known' fuel. The fact that a fuel has a certain label doesn't mean that it's identical to another drum with the same label. Ideally we want fuel from the same batch and that has been stored under the same conditions. This can mean buying in sufficient fuel to meet your test needs for an entire race season. Then at the beginning of the new season back to back tests would be done to compare performance with the 'old' and 'new' fuels.

The third variable is the dyno operator's test methodology. To get consistency in the numbers, the dyno operator has to be consistent in how he goes about his work.

Things like engine geometry on the test stand may not seem hugely important, but if

actual test procedure. The operator has to establish a set test sequence and not deviate

ROLLING ROAD PITFALLS

It is when working with a rolling road that the operator must be even more diligent not

extract more hp, when in reality

very basic tune-up.

The main problem centres on the fact that the rolling road relies on consistent

otherwise the tyre circumference will change

At this point you should be beginning to see glaring problems. If the measuring

rpm equals 100kph?

This last problem can be overcome if the dyno software is hooked up so as to

operator's eyesight and the accuracy of the car's tacho out of the loop

Tyre circumference is a different problem, and to minimise its influence on the

rotational speed increases, ie the tyre gets longer However tyre circumference decreases when we add more weight to the vehicle, or reduce speed In this situation

relationship that has been programmed into the dyno software

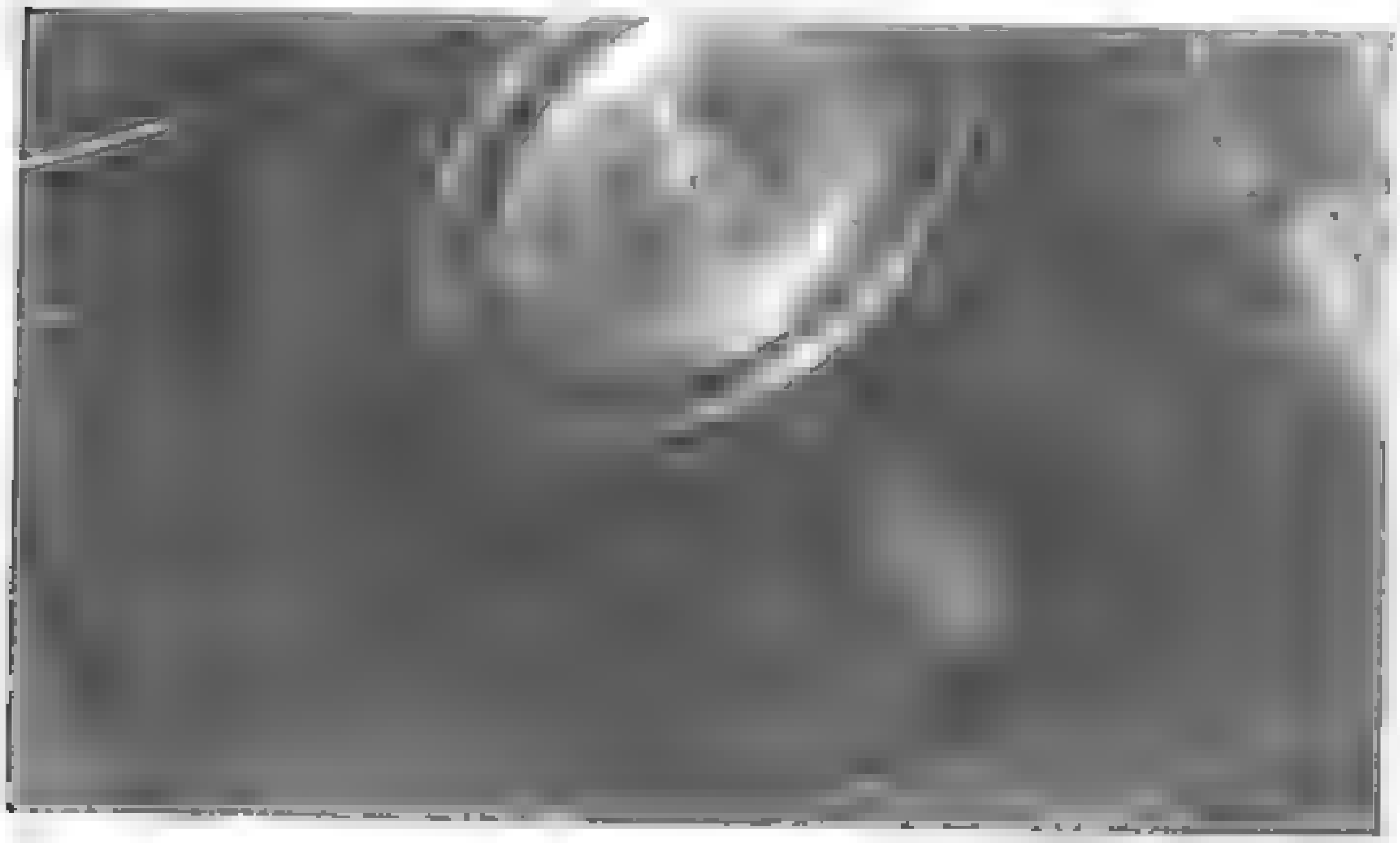
n equals garbage out'. Therefore the operator need the tyres on the rollers to bring them up to pressure, and then set the software to rpm for 100kph. If there is a wheelspin and the car has to be weighted down he will have to go back and reset the software at 100kph.

You, as the individual having your car dynoed, also have to be on the ball. to the problem on a rolling road drive tyre wedged down between two rollers, so with each revolution the tyre is compressed twice. As rotation speed increases so does the number of tyre ns. Consequently tyre pressure, on, tread width, tread depth etc will all be influencing friction the power graph. Therefore ideally for utmost consistency you should ted set of drive tyres that you use only for dyno tests. If you decide on th with your dyno tyres need to be sealed in plastic bags and stored out of light s to maintain their original characteristics as much as possible. You don't need new s, in fact the less tread the better. Obviously you need to correctly pressure tyres to identical pressure for each dyno test, and in this regard nitrogen is a more reliable gas than compressed air loaded with water vapour.

Over the years all sorts of ideas have been put forward to 'convert' hp at the wheels to hp at the flywheel. Some have the idea you can simply add a certain age – 15% is a popular number – to account for losses in the gearbox s, diff and tyres. Others prefer to add 20 or 30hp to the wheel hp numbers to make up for these losses. All such theories are dead wrong, but don't despair because a modern eddy current rolling road equipped with a bi-directional strain gauge can arrive at a pretty accurate figure – providing the methodology is correct.

e operator has to do at the end of the power run is pull the car out of gear

The dyno operator must do everything possible to eliminate test variables. With roll testing he has to try to keep the tyre circumference constant



and let it run down. Then by measuring the run-down rolling losses and adding them to the power graph we arrive at a graph that will come close to mirroring that from an engine dyno. However there are two provisos: first the operator has to select an acceleration rate to match the engine; and secondly, he has to ensure that there is no change in the load over the drive wheels between that applied during the power run and the run-down. If you doubt the reliability of this method, try doing a series of runs [illegible] would have been all over the shop. Generally the lower the gear the higher the whee[illegible] because of less tyre rotations during the power run. The run will also take less time so with less opportunity to put a lot of heat into the combustion chamber the engine may make more power.

At this point you may be wondering if an eddy current rolling road is so accurate when power runs are done properly, why are engine dynos the development tool of choice? Basically it all gets down to consistency and repeatability from one test to the next. With an engine dyno the test environment can be carefully controlled. This enables test results with variations of no worse than 1%, and in the very best engine shops working for well funded teams their back to back test consistency is closer to 0.5%. Even when you do all you can to eliminate variables, you can't do better than about 3% with a rolling road, although at certain points on the power curve it can be less than 2%. That's perfectly acceptable in many forms of motorsport, but when we are looking for small gains we need greater accuracy. There's also the aspect of convenience. With an engine dyno there is no time wasted installing the engine in a car prior to testing each new modification. And with the engine reasonably accessible while attached to the dyno many bits can be changed and tested quite quickly.

WHAT THE NUMBERS MEAN

[illegible] long connected to the crankshaft. If the engine moves this load through one revolution, work is being done; in this instance 628ft/lb (twisting force x revolutions x lever length x 2π). Power is the rate at which this work is being done, hence

$$\text{Power} = \text{work}\ \frac{(\text{torque x revolutions})}{\text{time}}$$

In the Imperial system, power is measured in pounds.feet per minute. However, these units are very small, so the unit we know as horsepower (hp) is the one used today. One horsepower equals 33,000lb ft per minute. This was calculated as a result of experiments carried out by James Watt, using strong dray horses. It is obvious

realising that power is the rate at which work is done, that two motors both producing 100lbf ft torque could have differing power outputs. In fact, if one motor lifted its 100lb load twice as quickly as the other, then it must be twice as powerful or have double the horsepower. Engine speed is measured in revolutions per minute and this is the time unit we use in calculating horsepower, so

$$hp = \frac{torque \times rpm}{5252}$$

Also

$$hp = \frac{bmep \times L \times rpm}{13,000}$$

where L = engine capacity in litres.

Earlier, I mentioned that high horsepower figures can be misleading. We can end up with a big power figure because the motor turns a lot of rpm, but unless it produces a higher torque output over a wide rpm range, lap times could be slower due to poorer acceleration or an increase in the number of gear changes required.

For this reason we have the measure of brake mean effective pressure (bmep), which gives a true indication of how effectively the motor is operating regardless of its capacity or its operating rpm. It is, in fact, a measure of the average cylinder pressures generated during all four engine strokes. We calculate bmep using the following formula:

$$bmep = \frac{hp \times 13,000}{L \times rpm}$$

$$bmep = \frac{lbf\ ft \times 2.475}{L}$$

where L = engine capacity in litres.

The highest bmep will occur at the point of maximum torque. In fact, this formula gives the true bmep only at that point. At other places on the horsepower or torque curves it is very close but not exactly true, but do not let this worry you.

The average motor runs at a bmep of around 140–165psi. A good road motor should run at 165–190psi. Rally engines will generally be in the 185–205 bracket and racing motors from 205–230psi. A few exceptional motors will run up to 240psi, naturally aspirated.

In standard form, motors such as the Alfa Romeo double overhead cam, the Lotus-Ford Twin Cam and the Ford BDA operate at around 170psi. Many motorcycles operate at a figure of about 165–180psi and superbikes are showing bmeps of 195psi, with a few nudging 200psi.

On paper, 175hp from a 1,300cc Ford BDA looks impressive, but when you [illegible] it all out it is rather less [illegible] [illegible] produced its torque at 7,200rpm and 175hp at 9,500rpm; the rev limit was 10,500rpm. The [illegible] was 194psi. A properly modified motor should have produced 207–210psi considering that it had four [illegible] [illegible] motors pump 205psi – and on a single four-barrel carburettor at that.

Another measure we use to analyse combustion efficiency is called brake specific fuel (bsf) [illegible] [illegible] combustion efficiency improves, up to the point where subsequent reduction in the bsf causes reductions in horsepower.

GEARING'S INFLUENCE ON PERFORMANCE

With the dyno session over and all the twiddling completed, do not go home and file the dyno sheets. [illegible] your gear set is suitable for the torque characteristics of your engine, and when you decide on a gear set work out your gear change points for maximum acceleration. In Table 14.2 I have listed the results of a 1600 Ford BDA dyno test. This is a strictly limited-budget club-level engine, a wet sump unit with a heat-treated cast-iron crank and stock rods that have been [illegible] [illegible] if possible.

Table 14.3 shows one method that can be used to check the suitability of various

Table 14.2 Dyno test of 1600cc Ford BDA

rpm	hp	torque (lbf ft)	bmep (psi)
4,500	94.4	110.2	170.5
5,000	111.3	116.9	180.8
5,500	130.4	124.5	192.6
6,000	154.5	135.2	209.2
6,500	169.2	136.7	211.5
7,000	178.9	134.2	207.6
7,500	190.1	133.1	205.9
7,750	190.9	129.4	200.2
8,000	183.2	120.3	186.1

gear sets. By calculating the 'step' between gears it is a simple matter to calculate what the engine rpm will be when you shift into the next gear. Then you can determine if the engine will be dropping out of the power band between gears. The T5 box and 'Rocket' gear set are very popular at club level, but how suitable are they? With the T5 the change from second to third gear with an 8,000rpm change point is going to see engine revs fall to about 5,500rpm, which is 500rpm out of the power band, but even worse, using a 7,750rpm redline puts the engine 700rpm out of the power band. The gap from third to fourth appears to be less of a problem, with a rev drop of around 2,000rpm, putting the engine 250rpm out of the power band. However, at this higher speed the car's frontal area and aerodynamic attachments are creating a lot more drag so the gap may be more of a problem than that between second and third. The Rocket set is more suitable, with the second-to-third change dropping engine rpm by about 1,900rpm, and the third-to-fourth change is dropping engine speed by 1,600rpm. By comparison, gearboxes used by racers with fat wallets, such as the ZF and Hollinger, appear to be an overkill, with the upchange in most gears dropping engine rpm by only 1,000–1,300rpm.

However, this assumption merely indicates just how inadequate a simple chart like Table 14.3 really is. A lot of people use charts like this, but I do not think that they

Table 14.3 Gearbox gear step and rpm step comparison

	Gear ratio	Gear step	rpm drop at redline rpm 6500	7000	7500	8000	8500	9000
T5	1st 2.95	+34.2%	2223	2394	2565	2736	2907	3078
	2nd 1.94	+30.9%	2008	2163	2318	2472	2627	2781
	3rd 1.34	+25.4%	1651	1778	1905	2034	2159	2286
	4th 1.00	+20%	1300	1400	1500	1600	1700	1800
	5th .80							
Rocket	1st 2.54	+34.3%	2230	2401	2573	2744	2916	3087
	2nd 1.67	+24.5%	1593	1715	1838	1960	2083	2205
	3rd 1.26	+20.6%	1339	1442	1545	1648	1751	1854
	4th 1.00							
ZF	1st 2.3	+30.4%	1976	2128	2280	2432	2584	2736
	2nd 1.6	+15.0%	975	1050	1125	1200	1275	1350
	3rd 1.36	+16.2%	1053	1134	1215	1296	1377	1458
	4th 1.14	+12.3%	800	861	923	984	1046	1107
	5th 1.00							
Hollinger	1st 2.57	+22.6%	1469	1582	1695	1808	1921	2034
	2nd 1.99	+19.1%	1242	1337	1433	1528	1624	1719
	3rd 1.61	+16.1%	1047	1127	1208	1288	1369	1449
	4th 1.35	+15.6%	1014	1092	1170	1248	1326	1404
	5th 1.14	+12.3%	800	861	923	984	1046	1107
	6th 1.00							

really tell us anything much about the suitability of a gearbox. Looking at gearing from the aspect of remaining within the engine's power band is all right for a road car, but for track work there are many other considerations. To some extent this illustrates why many racers look at tuning the engine to produce more hp when the opposition is showing them a clean pair of heels, when in reality there may be a lot more performance gain for money spent in areas such as brakes, suspension, weight, aerodynamics and gearing.

Certainly, more power will accelerate a car faster, but power is not the only factor in the equation. Vehicle weight obviously also figures prominently, as does gearing. Gearing is all about torque multiplication. If we have 100lbf ft torque at the crankshaft and 10:1 gearing, the twisting force at the axles will be 1,000lbf ft. That will cause the car to accelerate harder than an engine producing 10% more torque but with 8:1 gearing, or 880lbf ft (110 x 8 = 880) at the axles.

Looking at Tables 14.4 to 14.7 will help you to get a clearer idea of a frequently overlooked truth. Even though you are probably not interested in road speed in a race car, gearing figures need to be given a third dimension so that they can be related to the real world, and the only way that we can get the idea of what we are looking for in a gear set is to consider axle torque, road speed and engine speed together. Axle torque is found by multiplying engine torque by the overall gear ratio, while road speed can be worked out using this formula:

$$\text{Road speed} = \frac{\text{rpm} \times \text{tc}}{\text{gr} \times 1050}$$

Where tc = tyre circumference in inches (πd), and gr = overall gear ratio.

Checking Table 14.4 you can see the problem with the T5 gear set for this particular BDA engine. On the change from second to third we are dropping axle torque by

Table 14.4 Rear-axle torque and road speed with 5.14 differential and T5 gearbox

	1st 15.16		2nd 9.97		3rd 6.89		4th 5.14		5th 4.13	
rpm	*torque*	*speed*	*torque*	*speed*	*torque*	*speed*	*torque*	*speed*	*torque*	*speed*
[illegible]	[illegible]	21	1099	32	759	47	566	63	455	78
5,000	1772	24	1165	36	805	52	601	70	483	87
5,500	1887	26	1241	40	858	57	640	77	514	96
6,000	2050	28	1348	43	932	63	695	84	558	104
6,500	2072	31	1363	47	942	68	703	91	565	113
7,000	2034	33	1336	50	925	73	690	98	554	122
7,500	2018	36	1327	54	917	78	684	105	550	130
7,750	1962	37	1290	56	892	81	665	108	534	135
8,000	1824	38	1199	58	829	83	618	112	497	139
Torque drop*		40.6%		34.1%		24.6%		15.0%		

*Approximate torque drop at 7,750rpm red line

around 34% – from 1,290 at 7,750rpm in second down to about 850 at a little under 5,500rpm in third gear. In the higher gears the torque drop is less dramatic, but wind drag will contribute to slowing the rate of acceleration as road speed increases, so the figures are even less healthy than they appear on paper.

Table 14.5 Rear-axle torque and road speed with 4.7 differential and Rocket gearbox

	1st 11.94		2nd 7.83		3rd 5.90		4th 4.7	
rpm	*torque*	*speed*	*torque*	*speed*	*torque*	*speed*	*torque*	*speed*
[illegible]	[illegible]	[illegible]	863	41	650	55	518	69
5,000	1346	30	915	46	690	61	549	76
5,500	1487	33	975	50	735	67	585	84
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	798	73	635	92
6	1632	39	1070	[illegible]	807	79	642	99
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	792	85	631	107
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	785	91	626	115
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	763	94	608	118
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	710	97	565	122
Torque drop*	40.8%		24.0%		16.1%			

*Approximate torque drop at 7,750rpm redline

Table 14.6 Rear-axle torque and road speed with 4.7 differential and ZF gearbox

	1st 10.81		2nd 7.52		3rd 6.39		4th 5.36		5th 4.7	
rpm	*torque*	*speed*	*torque*	*speed*	*torque*	*speed*	*torque*	*speed*	*torque*	*speed*
4,500	1191	30	829	43	704	51	591	60	518	69
5,000	1264	33	879	48	747	56	627	67	549	76
5,500	1346	37	936	53	796	62	667	74	585	84
6,000	1462	40	1017	57	864	67	725	80	635	92
6,500	1478	43	1028	62	874	73	733	87	642	99
7,000	1451	47	1009	67	858	79	719	94	631	107
7,500	1439	50	1001	72	851	84	713	100	626	115
7,750	1393	52	973	74	827	87	694	104	608	118
8,000	1300	53	905	76	769	90	645	107	565	122
Torque drop*	34.2%		10.2%		11.4%		8.5%			

*Approximate torque drop at 7,750rpm redline

We see the same sort of problem with the Rocket gear set, big slumps in axle torque that, while not dropping the engine out of the power band, will handicap engine performance on the race circuit by not permitting the car to accelerate as quickly as is possible with a better selection of gears. However, when we look at the ZF and Hollinger boxes we see a quite different situation. With both gear sets there is only a slight drop in axle torque – about 10% – between gears.

Table 14.7 Rear-axle torque and road speed with 4.7 differential and Hollinger gearbox

	1st 12.08		2nd 9.35		3rd 7.57		4th 6.35		5th 5.36		6th 4.7	
rpm	*torq*	*spd*	*torq*	*spd*	*torq*	*spd*	*torq*	*spd*	*torq*	*spd*	*torq*	*spd*
4,500	1331	27	1030	35	834	43	700	51	591	60	518	69
5,000	1412	30	1093	38	885	48	742	57	627	67	549	76
5,500	1504	33	1164	42	942	53	791	62	667	74	585	84
6,000	1633	36	1264	46	1023	57	858	68	725	80	635	92
6,500	1651	39	1278	50	1035	62	868	74	733	87	642	[illegible]
7,000	1621	42	1255	54	1016	67	852	79	719	94	631	107
7,500	1608	45	1245	58	1008	72	845	85	713	100	626	115
7,750	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
8,000	1453	48	1125	61	911	76	764	90	645	107	565	122
Torque drop*	19.1%		14.9%		11.4%		10.8%		8.5%			

*Approximate torque drop at 7,75[illegible] rpm redline

[illegible]

[illegible] The Buick V6 was making the same hp as the Cosworth, about 750, but whereas the Buick was producing its maximum at 8,500rpm the Cossie was doing its thing at 12,500rpm. Hence the theory

that the Cosworth-equipped cars were quicker because the shorter gearing was getting them up to speed more rapidly off the turns.

When calculated out the theory does not hold up. At 8,500rpm the Buick is making 463.41lbf ft torque. Running a 3:1 axle, this becomes 1,390lbf ft at the axles. The Cosworth on the other hand is making 315.1lbf ft torque at 12,500rpm. Geared to run the same top speed as the Buick, the Cosworth will have to run a 4.41 axle (12,500 ÷ 8,500 x 3 = 4.41) which results in the same axle torque of 1,390lbf ft. Obviously lower gearing does not give the Cosworth an advantage.

It really gets back to engine design. At high rpm the Buick with rocker arms and push rods is struggling for reliability so it has to be geared so as to keep the engine together for a full 500 miles, which may mean limiting the rpm to 8,500 under race, as opposed to practice, conditions. The Cosworth on the other hand has no long-distance reliability problem with twin cams and small lightweight valves, so even when racing 500 miles it can be safely geared to run out to 13,000rpm. Clearly it is engine design, which allows the Cosworth to safely rev past maximum hp revs, together with the slightly shorter gearing that this permits, that gives the Cossie some advantage.

ASSESSING THE BMEP CURVE

This statement should not, however, be confused with the idea of making more power at high rpm. The problem for tuners is when a lot of rpm is being turned, but no more power is being gained. The 1,300cc Ford BDA mentioned earlier making 175hp at 9,500rpm is a good example of what I am getting at. This is an output that should have been achieved at 8,500rpm or perhaps 8,800 at the most. At 9,500rpm a 1,300 BDA should be showing 190–195hp. Why run an engine faster for no power gain when component life and reliability are reduced?

The time to run an engine harder is when you are at max bmep and you want more hp. For example, a 2-litre Opel running a bmep of 225psi at 8,000rpm is making 277hp. If it can be tuned to produce the same bmep at 8,250rpm, the power will go up to 286hp. This again illustrates why looking at bmep is so important; ordinary hp at dizzy rpm levels is a lot of pain for absolutely no gain.

Additionally, when we get the engine running at high bmep we want to tune it to give a flat bmep curve over a wide power band. Thus a four-cylinder, four valve, 2-litre motor running in a class rev-limited to 8,500rpm would be expected to pump close to 230psi at about 7,000 or 7,250rpm and not dip below 220psi all the way from 6,000 to 8,500rpm. Basically we do not want the bmep to fall more than 3–5% from maximum over a 2,500rpm range. Obviously to develop an engine to this level involves considerable expenditure of both time and money, but even with engines in a lesser state of tune the aim should always be to produce a nice broad and flat bmep curve. Therefore if engine design or budget restricts you to a realistic bmep of, say, 205psi, aim for that figure 1,200–1,500rpm lower than the maximum engine speed at which you can afford to operate and keep the bmep within 5% of 205psi, ie more than 194.7psi, over a 2,500rpm spread. For example, if maximum safe rpm is 7,500, try for a bmep peak at 6,000–6,250rpm with a bmep number over 195psi starting at 5,000 and going to 7,500rpm, or starting at 4,750 and going to 7,250rpm. Future development of the engine, if the rpm limit was to stay at

7,500, would centre around raising the bmep at the lower and upper ends of the curve to within 3% of the peak, thus we would be looking for a figure over 199psi. Such an improvement would see a power increase in a 2 litre engine of about 4hp between 5,000 and 7,500rpm.

In some types of competition, rallying for example, an even broader power band is desirable, and here the

valve, 2-litre engine wi

of something like 190psi at 4,500–5,000rpm and an hp peak at 7,000rpm.

GET AN ACCURATE TACHO

Obviously to get gear change points right and to avoid engine damage you need an accurate, easy-to-read tacho. The car manufacturer's stock unit is useless for

other problems that render it useless for competition, as we shall see.

Because we are running short gears, powerful engines and lightweight bodies, a

bigger problems.

A good-quality (expensive) race tacho will have these kinds of problems

incorporate a tell-tale and a shift light. A little more affordable are the Auto Meter

an Auto Meter) is the Elliott Clubman. It is a no-frills unit with no tell-tale or shift

 as I have seen a few 100–200rpm sl

SAFE ENGINE RPM

You are probably wondering just how fast it is safe to rev a motor, but of course there is no definite answer. The number of variables involved means that you must use an educated guess, or what is generally known as a rule of thumb. You must remember that engine wear increases dramatically at higher rpm, so a road motor should not be subjected to peak rpm operation continuously, unless you have the time and money for a rebuild every 7,000–10,000 miles.

The same type of reasoning applies in the case of the other types of motors where the rpm limit rule is applied. A rally motor operating at, or close to, the limit will require a rebuild at 1,200–4,000 miles; a road race engine every 300–1,500 miles; and a drag race motor every 1–4 meetings. The rebuild must include not only the replacement of worn parts, but also careful crack testing of components likely to suffer fatigue failure.

The rule of thumb we work to revolves around the mean piston speed in feet per minute. Many people feel that piston speed is of no consequence because of the large advances that have been made in modern-day metallurgy. In years gone by, this figure was used as a measure of an engine's likely wear rate — the higher the piston speed the faster the engine would wear out. I have found that working to an engine's piston speed is a surprisingly accurate means of avoiding blow-ups and general unreliability.

Mean piston speed is calculated using the following formula:

$$\text{Piston speed} = \frac{\text{stroke mm} \times \text{rpm}}{153} \quad \text{or} \quad \frac{\text{stroke inches} \times \text{rpm}}{6.024}$$

After you have calculated the rpm limit from Table 14.8 and the above formula, you may find that the limit imposed by the valve gear is lower. If this is the case, do not exceed the rpm limit of the valve train.

Table 14.8 Mean piston speed (fpm)

	Road	Rally	Road race	Drags
Standard cast iron crank and rods	3500	3500	3650	3800(B)
Standard forged crank and rods (A)	3500	4000	4200	4800(B)
Standard cast iron crank, special rods and heavy-duty main bearing caps	3500	4200	4350	4500(B)
Special forged crank, heavy-duty rods and main bearing caps (C)	3500–4000	4200–4400	4600–4800	5500–6500

Note: (A) applies to some standard high performance American V8 engines and many standard European and Japanese engines.

(B) applies to street machines used for occasional drag racing.

(C) most standard motorcycle engines also fit into this category due to the use of heavy-duty components in standard tune.

It is assumed that the engines in every category have balanced cranks, rods and pistons and that either forged or top-quality unslotted cast pistons are fitted.

Appendix

Table of Useful Equivalents

1 inch = 25.4mm
1 cubic inch = 16.387cc
1 horsepower = 0.7457 kilowatts
or 1.0139 PS
1 pound foot torque = 1.3558 Newton metres
or 0.13824 kg m
1 pound inch torque = 0.11298 Newton metres
1 psi = 6.89476 kilopascals
or 68.95 millibars
or 2.0345 inches of Mercury (Hg)
or 27.67 inches of water
°F = 1.8 x (°C + 32)
1 gallon (Imperial) = 160 fluid oz
or 4.546 litres
1 gallon (US) = 128 fluid oz
or 3.785 litres

1mm = 0.03937in
1 litre = 61.024cu in or 1,000cc
1 kilowatt = 1.341hp
1PS = 0.9863 horsepower
1 Newton metre = 0.7376 pound foot
1 kg m = 7.2336 pound foot

1kPA = 0.14504 psi
1 Bar = 14.5038 psi
or 100kPA

°C = (°F – 32) x 1.8
1 fluid oz (Imperial) = 28.4cc

1 fluid oz (US) = 29.57cc

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